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REPORT NO. RN-S-0428

PERFORMANCE EVALUATION
OF A MODIFIED NERVA TECHNOLOGY PUMP
IN LIQUID HYDROGEN AT ZERO NPSP

NERVA PROGRAM



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IN LIQUID HYDROGEN AT ZERO NPSP

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ABSTRACT

Cavitation tests were conducted with a modified NERVA technology pump to determine the performance changes resulting from vapor ingestion appearing as a variation of flow rate at the inlet of a fixed-geometry, fixed-speed pump.

Performance as a function of NPSH and flow rate, including favorable results at zero NPSP, and operational behavior of the positive NPSP pump are discussed. Back-vane performance is discussed in terms of static head coefficients.

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I. INTRODUCTION

The cavitation tests discussed in this report were performed in support of the NERVA turbopump program. The primary objectives of the test series were:

1. Determination of suction performance in liquid hydrogen as a function of flow rate.
2. Determination of optimum performance points at various levels of NPSP.
3. Acquisition of back-vane performance data for thrust prediction.

The turbopump currently under consideration for the NERVA engine must be able to operate at zero and positive suction pressures without incurring any significant cavitation losses. The subject test series was conducted to acquire data that can be employed for performance predictions of both separate inducers and centrifugal pumps with integral inducers operating at low values of NPSP. The tests were conducted during October 1966 at the Liquid Rocket Operations Test Area of the Aerojet-General Corporation in Sacramento, California, under Contract SNP-1 of the Space Nuclear Propulsion Office, Cleveland, Ohio.

II. SUMMARY

Cavitation tests were conducted with a modified NERVA technology turbopump to determine suction performance in liquid hydrogen as a function of pump flow rate. The pump incorporated a centrifugal impeller with integral inducer and a vaned diffuser discharging into a double-volute single-exit casing. The rotor had an inlet tip blade angle of 7° , an inlet tip diameter of 6.95 in., and a

discharge blade angle of 90° . The pump was tested at a constant inducer tip speed of 570 ft/sec and at seven inlet flow coefficients ranging from 0.063 to 0.088. Suction pressure was varied as a function of pressure rise.

A 24-in.-dia "PSOV vessel" containing the propellant shutoff valve was located 60 in. upstream of the inducer inlet to simulate outflow velocities of 3 to 4 ft/sec anticipated at the tank bottom of a nuclear propulsion module. Suction performance results presented in this report are therefore related to fluid conditions that prevailed at this tank-bottom simulation station as well as in the suction line.

Stable pump performance was achieved at total pressures equal to vapor pressure at the tank bottom. The tests demonstrated the capability of the pump to operate with significant amounts of ingested vapor (20 to 30% by volume) at values of NPSH equal to or less than the inducer inlet velocity head.

III. DISCUSSION

A. TEST CONFIGURATION

Tests were conducted with a modified version of an NRX/EST (Mark III, Mod 4) turbopump. Requirements for testing at inlet flow coefficients below or equal to the nominal inlet flow value of 0.07 necessitated a hardware modification to shift diffuser stall to a lower flow coefficient. This correction was accomplished with a narrower diffuser port width (0.400 vs 0.550 in.) and a narrower diffuser inlet vane passage height (0.660 vs 0.800 in.) and by decreasing the impeller discharge diameter from 12.25 to 10.4 in. The reduction in impeller diameter resulted in the removal of the short radial vanes, thus causing a decrease in the total number of vanes from 48 to 24. The modification also resulted in a very large (1.14 in.) radial clearance between impeller and diffuser. Figure 1

shows the geometrical relationship of the major pump components. Parameters describing the pump configuration are listed in Table 1.

The pump was driven by a two-stage axial-flow turbine. Power was transmitted through the shaft supported by bearings located between the two rotating elements.

B. TEST FACILITY AND CONTROLS

Testing was conducted on Test Stand H-6 at Aerojet's Sacramento Plant. The pumping fluid (LH_2) was supplied from a 105,000-gal vacuum-jacketed run tank through a 24-in.-dia FSOV vessel containing the propellant shutoff valve (PSCV) and a 60-in.-long, 8-in.-dia, vertical pipe to the pump inlet. The turbine was driven with warm gaseous hydrogen from the facility heater. A schematic of the test setup is shown in Figure 2.

Pump speed was controlled automatically by the turbine power-control valve (TPCV) operating in closed loop with the speed pickup. The system was capable of controlling the speed within 1% of the preselected value of 22,000 rpm.

The pump flow rate was initially controlled manually, and subsequently maintained with closed loop Q/N control, with the flow control valve and measured upstream flow rate. During cavitation tests, suction pressure was varied as a function of pump pressure rise by controlling tank pressurization and vent valves.

C. INSTRUMENTATION

Parameters measured for suction performance evaluation included pressures and temperatures in the tank bottom (Station 75), in the pump inlet line (Station 28), and at the pump discharge. Station locations are shown in Figure 2.

Platinum resistance probes (Rosemount RTTs) were used to measure propellant temperatures in the pump suction and discharge line. The two-sigma system accuracy was estimated to be within 0.069°F , resulting in a possible vapor-pressure error of 0.2 psi. Low- and high-range pressure transducers (0 to 50 and 0 to 100 psi) were used at each of the reference stations (Stations 28 and 75) to increase the accuracy of the measurements. The one-sigma system accuracy was estimated to be within 0.3% of the full-range value. Flowmeter accuracy was estimated to be within 1% of the flow rates tested.

Fluid temperature at the back side of the impeller was measured with chromel-constantan thermocouples. An on-stand nitrogen bath, whose temperature was monitored with an RTT, was used as a reference. Signal conditioning was achieved with a cascade of two Dana amplifiers, each set at a gain of 100 to obtain an overall gain of 10,000. The obtained measurement accuracy was estimated to be better than the full-scale variation of $\pm 1.5^{\circ}\text{F}$ determined in a similar application.⁽¹⁾*

D. TEST SUMMARY AND PROCEDURES

Eight tests were conducted. One noncavitating head-capacity test was conducted to determine the noncavitating performance and the stall limit of the modified pump and was followed by seven cavitation tests at inlet flow coefficients ranging from 0.063 to 0.088. The tests were performed in four runs (two tests per run) of approximately 15-min duration. The cavitation tests were conducted at constant speed and constant flow rates according to the sequence indicated in Table 2.

*Superior numerals refer to items in the List of References, unless obviously intended for another purpose, such as to indicate an exponent.

All tests were performed at 22,000 rpm. The head-flow test was conducted at suction pressures ranging from 60 to 80 psia, whereas subsequent cavitation tests were started at suction pressures of approximately 50 psia. During cavitation testing, suction pressure was reduced in specified increments to the level where incipient cavitation was indicated by a drop in pump pressure rise. When this level was reached, suction pressure was controlled as a function of pump pressure rise.

Fluid temperature was maintained between 37.5 and 38.5°R during the entire series of tests.

E. PERFORMANCE PREDICTIONS

1. Noncavitating Performance

Performance of the low-flow pump was estimated from the measured performance of the unmodified NRX/EST pump. For constant speed, the head rise of the modified pump was obtained from

$$\Delta H_{\text{mod}} = \left[\frac{D_{2 \text{ mod}}}{D_{2 \text{ NRX/EST}}} \right]^2 \frac{\eta_{\text{mod}}}{\eta_{\text{NRX/EST}}} \left(\Delta H_{\text{NRX/EST}} \right),$$

where

$D_{2 \text{ mod}}$ = impeller discharge diameter of modified pump = 10.40 in.,

$D_{2 \text{ NRX/EST}}$ = impeller discharge diameter of NRX/EST pump = 12.25 in.,

η_{mod} = slip factor of modified pump = 0.918 ($n = 24$),

$\eta_{\text{NRX/EST}}$ = slip factor of NRX/EST pump = 0.959 ($n = 48$).

Obtained head rise and efficiency (from NRX/EST pump) were degraded by 5% to account for losses resulting from the unusually large impeller-to-diffuser radial clearance.

The stall point was estimated from the ratio of throat areas, diameters, and slip coefficients, by assuming the diffuser would stall at approximately the same stall incidence angle as that of the NRX/EST pump.

$$\left(\frac{Q}{N}\right)_{\text{Stall}}^{\text{mod}} \approx \frac{(\frac{D_2}{D_1})^2 \left(\frac{A}{A_1}\right)^{\text{mod}}}{\left(\frac{D_2}{D_1}\right)^2 \left(\frac{A}{A_1}\right)^{\text{NRX/EST}}} \left(\frac{Q}{N}\right)_{\text{Stall}}^{\text{NRX/EST}} = 0.121,$$

where A = diffuser throat area = 3.52 sq in. for NRX/EST (2.21 sq in. for modified pump).

Predicted pump head rise and efficiency are depicted in Figure 3.

2. Suction Performance

The net positive suction head (NPSH) at 3% head loss was predicted by using the approximation

$$\text{NPSH} = 3 \frac{C_m^2}{2g}.$$

For nominal conditions, where $\phi_{IT} = 0.071$ and $C_m = 47$ ft/sec, the above expression yielded a NPSH value of 104 ft.

F. TEST RESULTS

1. Noncavitating Performance

Test results plotted as normalized head rise and efficiency are presented in Figure 3, together with the characteristic curves of the unmodified pump. The modification successfully shifted the pump operating range toward the desired values of Q/N , but greater-than-anticipated losses occurred as a result of the large radial clearance between the impeller and diffuser. The greater-than-expected negative slope of the head flow curve, which is similar to

the characteristic curve of a backward-curved-blade impeller, was attributed to the greatly reduced impeller radial discharge solidity resulting from the decrease in impeller discharge diameter and removal of the short radial blades.

Figures 4 and 5 show the noncavitating performance as a function of discharge and inlet flow coefficients. Diffuser stall occurred at the predicted flow rate, resulting in a stall margin of 50% at the nominal flow rate based on ϕ_{IT} of 0.071. All noncavitating data were acquired at suction pressures ranging from 60 to 80 psia.

2. Cavitation Performance

Excursions in suction pressure from 30 psi above vapor pressure down to vapor pressure and back to initial pressure were conducted successfully at inlet flow coefficients of 0.071, 0.078, 0.084, and 0.0864. Cavitation tests at flow coefficients of 0.063, 0.067, and 0.088 yielded valuable data, although vapor pressure in the suction line could not be reached because of premature test termination caused by propellant depletion. These three tests were conducted during the latter portion of runs 2, 3, and 4. An attempt to conduct the last test at an inlet flow coefficient of 0.060 is reported in Paragraph III.F.3.

Time traces of variables describing inlet conditions and performance are shown on Figures 6 through 9 for the four tests during which vapor pressure was reached in the suction line. Characteristics typical in these tests are reported in the following paragraphs.

The change in state from single-phase to two-phase flow was evidenced by a drop in fluid temperature in the suction line. This change occurred while the bulk temperature remained constant and is best indicated by

the vapor pressures P_{Vap}^{-28} and P_{Vap}^{-75} in Figures 6 (Sheet 1) and 9 (Sheet 1). In Tests 3 and 5, boiling occurred also in the PSOV vessel and is indicated by a small drop in P_{Vap}^{-75} in Figures 7 (Sheet 1) and 8 (Sheet 1).

The temperature TFS-28 or the related pressure P_{Vap}^{-28} dropped off before the static pressure PFS-28 reached vapor pressure. Apparently, local boiling started gradually and, presumably, first around protrusions. Vapor is believed to have formed first at the submerged temperature probe, causing measurements to indicate local temperature lower than the average fluid temperature in the line. Accordingly, P_{Vap}^{-28} was a local vapor pressure of lower magnitude, reflecting fluid boiling at the temperature probe.

The gas volume expressed in percent of the total fluid volume plotted in Sheet 2 of Figures 6 through 9 was estimated from the measured temperature drop at Station 28 and from the basic heat-balance equation:

$$\dot{w}_{\text{gas}} L = \dot{w}_{\text{liq}} C_p \Delta T,$$

where

$\dot{w}_{\text{gas}}$ = weight flow rate of gas (lb/sec),

$\dot{w}_{\text{liq}}$ = weight flow rate of liquid (lb/sec),

L = latent heat of vaporization (Btu/lb),

C_p = specific heat of saturated liquid at constant pressure (Btu/lb-°R),

ΔT = temperature drop (°R).

Volume ratios were calculated from densities obtained from fluid-property subroutines as functions of the fluid quality $\dot{w}_{\text{gas}}/(\dot{w}_{\text{liq}} + \dot{w}_{\text{gas}})$. With regard to the previously mentioned fluid condition at the temperature probe, the gas fractions calculated for Station 28 might be overestimated but are believed to be realistic when referred to the pump inlet because the vapor content increases with line length.

Static suction pressures lower than the local vapor pressure by as much as 0.4 to 0.5 psi were measured in the suction line. In many instances, the measured differences exceeded the estimated accuracy of the two measuring systems combined. From previous experience with the type of instrumentation used, the total systems error should not have exceeded 0.25 psi. As previously stated, vapor pressure was based on the temperature measured by the shielded RTT at Station 28, whereas the static pressure was measured at the wall approximately 1.5 in. from the temperature probe. Although measurements from such temperature probes are more indicative of total temperatures than static temperatures, the difference between them is still too small (0.024°R) to relate the measured effect to instrumentation accuracy alone. A correction, to account for the possibility that boiling around the temperature probe could have caused the measurements to indicate lower-than-average line temperatures, would only enhance the difference between the two pressures. Consequently, the cause of this phenomenon must be attributed either to a non-equilibrium condition of the fluid (superheated fluid) or to some undetermined nonuniform boiling in the line resulting in lower fluid temperatures at the wall. As a result of this phenomenon, values of NPSH lower than the velocity head were measured in the suction line; and similarly, negative values of NPSP were determined at Station 75 in the PSOV vessel during Tests 3, 5, and 7.

Repeatability of conditions and of the effects encountered was excellent. During Test 5 (Figure 8), three excursions with intermediate recovery to positive NPSP were conducted, each showing substantially the same result.

Moderate flow variations during deep cavitation reflect the capability of a manual flow-control system based on inlet flowmeter measurements. The inlet and the discharge flowmeters showed good agreement throughout the duration of the tests except during periods of heavy pump cavitation. During such periods, the inlet flowmeter started to overspeed because of the increase in volumetric flow resulting from the change in fluid density in the cavitating meter, causing the operator to readjust the inlet valve. As a result, a drop in weight flow rate can be noticed at each occurrence. All plotted values of weight flow are based on discharge flowmeter measurements.

Performance and inlet conditions for points of minimum NPSH and maximum drop in performance are summarized in Table 3. All data that are indicated reflect transient conditions.

Figure 10 presents the suction performance characteristics of the pump in terms of overall head coefficient as a function of the ratio of NPSH to inlet axial velocity head for a family of flow coefficients. The values of $NPSH/C_m^2/2g$ of less than one denote negative static NPSH and reflect the phenomena discussed in the preceding section (III.F.2). The data indicate that the pump can be operated at a value of NPSH equivalent to three times the inlet velocity head but would incur a performance loss of approximately 9%. The predicted loss at that value was 3%.

The same results are presented in Figures 11 and 12 as functions of the cavitation parameters $\mathcal{J}$, $(NPSH/U^2/2g)$ and $K \left(\frac{NPSH - C_m^2/2g}{W_{IT}^2/2g} \right)$. The negative values of K result from measured NPSH values smaller than the inlet velocity head.

Figure 13 shows the performance drop expressed as the ratio of cavitating to noncavitating head rise as a function of normalized NPSH based on line measurements for selected flow coefficients. The same results are plotted in Figure 14 as functions of NPSP measured in the PSOV vessel, with several data points at zero NPSP. The lower values of head along the zero-NPSP line represent increased amounts of ingested vapor.

Figure 15 represents changes in performance as functions of the suction specific speed. The fact that head degradation started at relatively low values of S (10,000 to 20,000) but that the pump was still capable of operating at values of S exceeding 200,000 was unexpected.

From data presented in Figures 13 and 14, it is apparent that the flow rate at which minimum head loss occurred was dependent upon NPSH. This fact is well indicated by the lines of constant flow rates that intersect each other at various levels of NPSH. The same trend is discernible in Figure 16, which depicts the cavitation parameter ϕ as a function of the inlet tip flow coefficient for 3, 5, and 7% head loss. The curves indicate an improvement in relative performance with increasing flow at lower values of ϕ . Figure 17 presents head loss as a function of flow rate or incidence-to-blade-angle ratio during conditions when the total pressure in the PSOV vessel reached vapor pressure, simulating a zero-NPSP tank condition. The curve shows good correlation with Lewis Research Center (LeRC) inducer data, indicating higher losses at low values of incidence. Previous LeRC tests had indicated that a wide range of excursions to high incidence was possible with virtually no change of performance.

Figure 18 shows the cavitation parameter K at 3% head loss as a function of the inlet tip flow coefficient, together with the region indicating Aerojet experience with pumps tested in water. K falls within the defined

cavitation performance envelope. The higher-than-anticipated curve values reflect the typical suction-pressure sensitivity of the pump at high levels of NPSH.

3. Operational Characteristics

Operational considerations such as speed control and pressure-oscillation amplifications during suction-pressure excursions to zero NPSP are of major interest from an engine-design standpoint. The following observations were made:

a. Pump efficiency, herein expressed as the ratio of pump hydraulic head rise to pump-fluid enthalpy rise, was observed to follow pump head rise closely. The ratio of head rise to pump efficiency remained constant within 2% up to practical head-loss limits of about 5%. Even at the higher values of head loss (20 to 50%), the head-rise-to-efficiency ratio remained constant within 5%. As a result, when pump cavitation occurred and pressure rise decreased, the tendency of the pump to overspeed when supplied with fixed turbine power was practically offset by a loss in pump efficiency, the loss being sufficient to produce essentially a fixed pump power requirement. In general, however, the percentage drop in pump head rise always exceeded the drop in efficiency. This effect has been commonly observed with other centrifugal pumps tested at Aerojet.

b. Oscillation Amplification

During these tests, flush-mounted high-frequency pressure transducers were located immediately upstream and downstream of the pump. However, during the analysis of the data, it was concluded that accurate determination of amplitudes from the data was unreliable because of noise effects. The comparison of relative conditions, however, was of interest.

Low-frequency suction and discharge pressure oscillations were recorded during the noncavitating head-capacity test and were found to vary in amplitude as a function of flow rate. The frequency of these oscillations was about 200 cps, with amplitudes about 40% greater at 85% of the nominal inducer flow rate ($\phi_{IT} = 0.071$) than at the highest and lowest flow rates. These oscillations are considered a basic characteristic of this pump.

At inlet tip flow coefficients above the nominal value of 0.071, low-frequency oscillations (approximately 200 cps) seemed to disappear during deep cavitation at NPSP values of less than 2 psi. At inlet flow coefficients below the nominal value ($\phi_{IT} < 0.071$), however, oscillations continued to increase in amplitude as suction pressure decreased. A cavitation test at 85% of the nominal flow rate ($\phi_{IT} = 0.060$) was aborted after a 7-psi NPSP was reached in the suction line because of increasing instability or surging. During this attempt, a threefold increase in amplitudes in both the suction and the discharge line was evidenced, reflecting nearly constant amplification of oscillations. In summary, the tests showed that pump operation at very low values of NPSP was very stable, but that dynamic instability or system surging at low flow effected a large reduction in the operating range.

4. Back-Vane Performance

Back-vane performance was evaluated to acquire non-dimensional data for estimating the pressure profiles of similar rotor designs. Accurate assessment of pressure profiles and related pressure forces during the design of a pump is essential in determining realistic bearing loads.

The back-vanes operated as a separate pumping system with a static discharge head controlled by the discharge pressure of the main pump impeller. The resulting inlet or cavity pressure depended on the pressure-generating ability of the back-vanes as well as the pump suction pressure. Back-vane flow was tapped off the pump discharge line and directed through a flow-measuring section and a flow-control orifice into the impeller rear cavity. All internal flow passages leading into the cavity were blocked off. Seal leakage into the cavity was estimated to be very small and was therefore neglected in evaluating flow sensitivity. Pressure-tap and temperature probe locations are shown in Figure 1.

Back-vane performance is expressed in terms of a static head coefficient defined as:

$$\psi = 144 \frac{\Delta P \text{ g}}{\rho U_T^2} = \frac{6.09 (10^7) \Delta P}{N^2 R_T^2},$$

where

U_T = peripheral velocity at back-vane discharge (ft/sec),

R_T = back-vane discharge radius (in.),

ρ = average value of fluid density (lb/ft³),

P = pressure rise across back-vanes, $P_{BVT} - P_{BV-1}$ (psi).

The highly-temperature-sensitive density of the warmer back-vane fluid (LH₂ near critical point) was determined from measured static pressures and temperature profiles established from the three temperatures T_{BV-1} , T_{BV-2} , and T_{BV-3} . At very low back-vane flow rates (2.4 lb/sec maximum), frictional heating was expected to cause a measurable temperature gradient, with a maximum temperature near the back-vane discharge. However,

on the contrary, the fluid temperature near the discharge of the back-vanes was actually lower than the temperature of the fluid in the cavity. The annular cavity formed by the unusually large diffuser-to-impeller radial clearance (See Figure 1) is believed to have encouraged considerable backflow and fluid mixing in the back-vanes. Backflow apparently was effective enough to make the temperature profile practically insensitive to changes in input flow. At zero back-vane flow (Test No. 8), the temperature difference ($T_{BV-2} - T_{BV-1}$) was predominantly smaller but only by a small fraction of a degree. Variations in measurements were within the accuracy of the measuring system.

Typical back-vane fluid temperatures recorded during non-cavitating test No. 1 are plotted as a function of flow in Figure 19. These temperatures follow the trend indicated by the fluid temperature in the pump discharge line and, thus, are also dependent on pump head rise and efficiency.

The temperature profile in Figure 19 (lower graph) shows the change in fluid temperature with respect to the cavity temperature T_{BV-1} . As indicated by the solid line, it is believed that the temperature tends to rise initially before dropping off as the back-vane discharge is approached. This temperature rise would be greater if backflow were forestalled to a greater extent - for example, by a conventional, considerably narrower flow passage at the impeller tip or by a back-vane diffuser such as employed on the M-1 oxidizer turbopump².

During pump cavitation, all static back-vane pressures dropped with the pump discharge pressure until the lowest pressure reached vapor pressure and the fluid started to boil in the cavity (Figure 20). A continued drop in pressures caused the vapor-pressure level to extend toward larger radii, thereby decreasing the pressure gradient across the back-vanes. The cavity

temperature dropped almost 6°R during the boiling phase of Test No. 7, while the pump discharge temperature remained practically constant. The depression in temperature or drop in vapor pressure resulted from vaporization cooling of the fluid. It is because of this temperature depression that the back-vane temperature profile represented by the dotted line in the lower graph of Figure 19 indicates a rise in fluid temperature during pump cavitation.

Back-vane performance is shown in Figure 21 in terms of static head coefficients as a function of radial distance from the shaft center line. Data were acquired at shutoff as well as at a nominal back-vane flow rate of 2.0 to 2.4 lb/sec. The change in flow rate from nominal to shutoff resulted in a 6% increase in static head rise. The identical slopes of the two profiles along the vaned section indicate that the back vanes were virtually insensitive to flow. The increase in static head rise (cavity to back-vane discharge) resulted from the increased head-generating ability of the vaneless section at zero flow. The vaneless section can be compared to a section of a suction line in which the fluid is affected by prerotation. The maximum pressure gradient is achieved in both sections at shutoff. It is generally recognized that the pressure gradient generated by a friction-induced vortex falls off rapidly as flow is produced, whereas a radial-vaned impeller tends to maintain a relatively stable discharge head at low flow rates. The flow sensitivity of back vanes was analyzed in greater depth in water tests performed on the M-1 oxidizer scale pump³.

A useful expression for analyzing the performance of various back-vane sections is the static head coefficient based on section radii, defined as:

$$\psi_{1-2} = \frac{1.89 (10)^6 g (P_2 - P_1)}{\rho N^2 (R_2^2 - R_1^2)} .$$

Static head coefficients for sections defined by pressure-tap locations (Figure 1) are shown in Table 4. The average value for the entire vaned section ($\psi_{BV\ 2-6}$) is 0.46. This value is somewhat lower than the equivalent K value of 1 claimed by Pratt & Whitney Aircraft⁴, which corresponds to a static head coefficient of 0.50. The clearance-to-vane-height ratio of the back vanes was $0.045/0.200 = 0.225$ and lies within the values tested in Reference 4. In general, the back-vane performance of this pump was about 5% better than that of the M-1 oxidizer scale pump tested with water (Reference 3). A minor performance drop near the discharge was attributed to a lower vane loading resulting from the large radial clearance between back-vane discharge and housing.

IV. CONCLUSIONS

The positive-NPSP pump was capable of operation at simulated zero-NPSP tank conditions or at measured NPSH values equal to or less than the inlet velocity head and was capable of ingesting large amounts of vapor (20 to 30 vol.%). Performance losses under these conditions, however, were unacceptable for a flight system and are attributed to excessive tip speed and relatively high inlet velocities.

Head loss of the pump was 9% at an NPSH value equivalent to three times the inlet velocity head at the nominal inlet tip flow coefficient of 0.071. This loss was greater than estimated and is attributed to the typical suction-pressure sensitivity of this pump at NPSP levels of up to 35 psi.

The optimum flow rate for minimum performance loss appeared to vary, depending on NPSH. Higher losses at higher flow rates generally occurred at high values of NPSH but also occurred under conditions of zero NPSP in the PSOV vessel. Performance changes at zero NPSP, however, did not vary strongly with flow. In general, performance was better than expected at the higher flow rates.

The range of stable pump operation was greatly reduced at NFSP values below 8 psi because of dynamic instability or surging. Above the critical flow rate ($\phi_{IT} \approx 0.063$), the pump operated very smoothly even when ingesting large amounts of vapor (20 to 30 vol.%). In fact, the low-frequency oscillations appeared to disappear in regions of deep cavitation.

The cavitation test series demonstrated that speed can be maintained even with a turbine drive because of the nearly constant power requirement of the pump.

Fluid temperature across the impeller back-vane decreased with increasing radius. When the fluid in the rear cavity started to boil as suction pressure was lowered, the temperature in the rear cavity dropped further, reversing the temperature gradient across the back vanes. The unexpected decrease in fluid temperature with increasing radius was attributed to increased backflow encouraged by the large impeller-to-diffuser radial clearance.

Back-vane static head coefficients for various radial sections varied between 0.44 and 0.48 and agreed quite well with those of other Aerojet pumps tested in water.

TABLE 1

PUMP CONFIGURATION
(NERVA Low-Flow Technology Pump,
Assembly FN 1117889-9)

Flow Areas

Suction line, sq in.	48.65
Pump inlet (no vane blockage), sq in.	35.26
PSOV vessel, sq in.	452.0
Discharge line, sq in.	17.72

Impeller FN 1117888-1

Inlet tip diameter, in.	6.950
Discharge tip diameter, in.	10.40
Hub diameter, inducer inlet, in.	1.846
Blade inlet tip angle, degrees	7.0
Blade inlet cant angle (measured from normal to the axis of rotation), degrees	0
Blade discharge angle, degrees	90
Inducer type	Cambered
Number of full vanes	3
Number of long partial vanes	3
Number of medium partial vanes	6
Number of short partial vanes	12
Impeller discharge vane height, in.	0.565
Impeller discharge vane blockage, %	13.1
Impeller inlet vane blockage, %	10.0
Back-vane tip diameter, in.	10.40
Back-vane inlet diameter, in.	6.20
Back-vane tip height, in.	0.200
Number of back vanes	48

Diffuser FN 1117885-9

Diffuser inlet angle, degrees	8°30'
Diffuser inlet port width, in.	0.400
Diffuser inlet vane passage height, in.	0.660
Diffuser base circle, in.	12.96
Number of diffuser vanes	8

TABLE 2
TEST SUMMARY

<u>Test No.</u>	<u>Run No.</u>	<u>Type Test</u>	<u>Speed, rpm</u>	<u>Q/N</u>	<u>Inlet Flow Coeff (ϕ_{IT})</u>	<u>Percent of Q₁ Nom</u>	<u>Axial Inlet Velocity (C_m), ft/sec</u>	<u>c/β_{IT}</u>
1	1	H-Q	22000	0.117-0.333	0.0351-0.10	49.5-141	23.5-67	0.71-0.18
2	1	Cav	22000	0.280	0.084	118.5	56	0.314
3	2	Cav	22000	0.236	0.0708	100	47.3	0.42
4	2	Cav	22000	0.212	0.0636	90	42.5	0.48
5	3	Cav	22000	0.260	0.078	110	52	0.36
6	3	Cav	22000	0.295	0.0885	125	59	0.28
7	4	Cav	22000	0.288	0.0864	122	57.6	0.295
8	4	Cav	22000	0.224	0.0673	95	45	0.45

TABLE 3

CAVITATION-TEST-DATA SUMMARY

(Performance and fluid condition at lowest values of NPSP)

Inlet flow coefficient (O_{IT})	0.084	0.0708	0.078	0.0864
Fluid bulk temperature, °R	37.65	37.7	37.9	38.35
<u>Lowest NPSP Values Reached, psi</u>				
NPSP-line = (P_s tot-line) - (P_{vap} -line)	0.4	0.15	0.17	0.3
NPSP-tank = (P_s tot-tank) - (P_{vap} -tank)	0.5	-0.2	-0.1	-0.1
<u>Maximum Values Reached:</u>				
Head loss, % of noncavitating head	13	29	27	49
Decrease of Efficiency, % of noncavitating efficiency	10.8	27.2	21.7	44.6
Ingested gas, vol. % of mixture	20	32.5	24	21.5
Suction specific speed, $S = \frac{NQ^{1/2}}{NPSH^{3/4}}$	204,400	304,500	328,000	242,000
Based upon NPSH in line cavitation parameter, $K = \frac{NPSH - C_m^2 l/2g}{w_{IT}^2/2g}$	-0.0016	-0.0020	-0.0024	-0.0020

TABLE 4
BACK-VANE PERFORMANCE

Back-vane static head coefficients defined as:

$$\psi_{BV2-1} = \frac{1.89 (10)^6 g (P_2 - P_1)}{\rho_{N_2} (R_2^2 - R_1^2)}$$

<u>Test Run</u>	<u>Time, Sec</u>	<u>$\dot{W}_{BV}$, lb/sec</u>	<u>BV1-2</u>	<u>BV2-4</u>	<u>BV4-6</u>	<u>BV2-6</u>
1	168	2.40	0.055	0.480	0.446	0.463
1	194	2.32	0.069	0.478	0.446	0.457
1	303	2.33	0.077	0.481	0.443	0.460
1	350	2.13	0.082	0.478	0.446	0.450
4	348	2.2	0.019*	0.442*	0.461	0.456
4	728	0	0.40	0.477	0.448	0.462
4	790	0	0.45	0.490	0.453	0.470

*Low values resulted from cavitation at back-vane inlet ($P_{BV-1} = P_{VAP}$).

NERVA LOW FLOW TECHNOLOGY TURBOPUMP STATIC PRESSURE TAP & TEMPERATURE PROBE LOCATIONS

SYMBOL	R (IN)
PBV 1	2.84
PBV 2	3.05
PBV 3	3.55
PBV 4	4.05
PBV 5	4.55
PBV 6	5.05
PBV 7	5.55
TBV 1	2.70
TBV 2	5.00
TBV 3	5.35

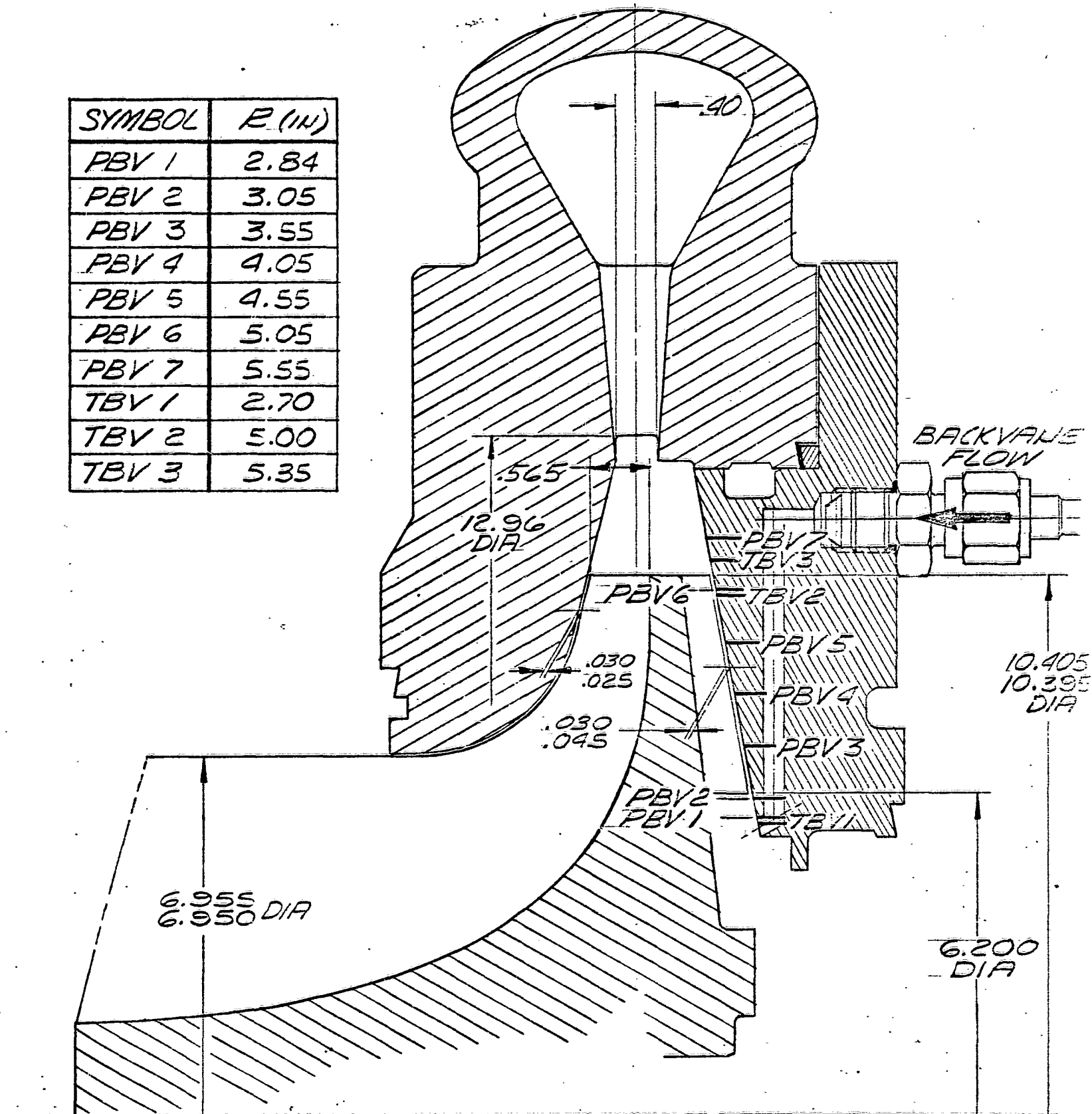


Figure 1
23

TEST SET UP SCHEMATIC
TEST STAND H-6

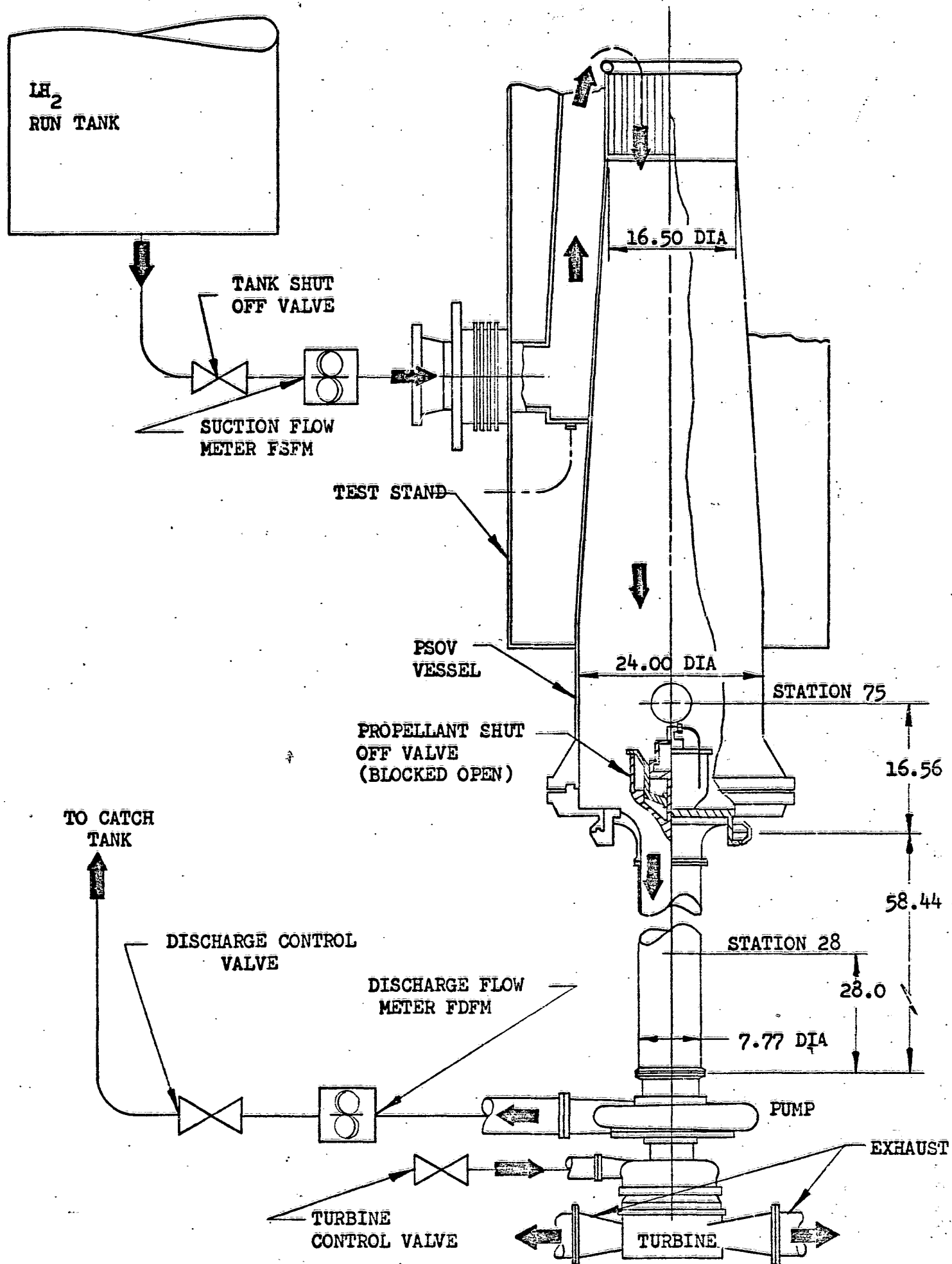


Figure 2
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NERVA TECHNOLOGY PUMPS COMPARISON OF NON-CAVITATING PUMP PERFORMANCE

TEST SERIES 0914-B01-0P-001

$N = 22,000 \text{ rpm}$

$P_s = 50-70 \text{ psia}$

FLUID: LH_2 @ 38°R .

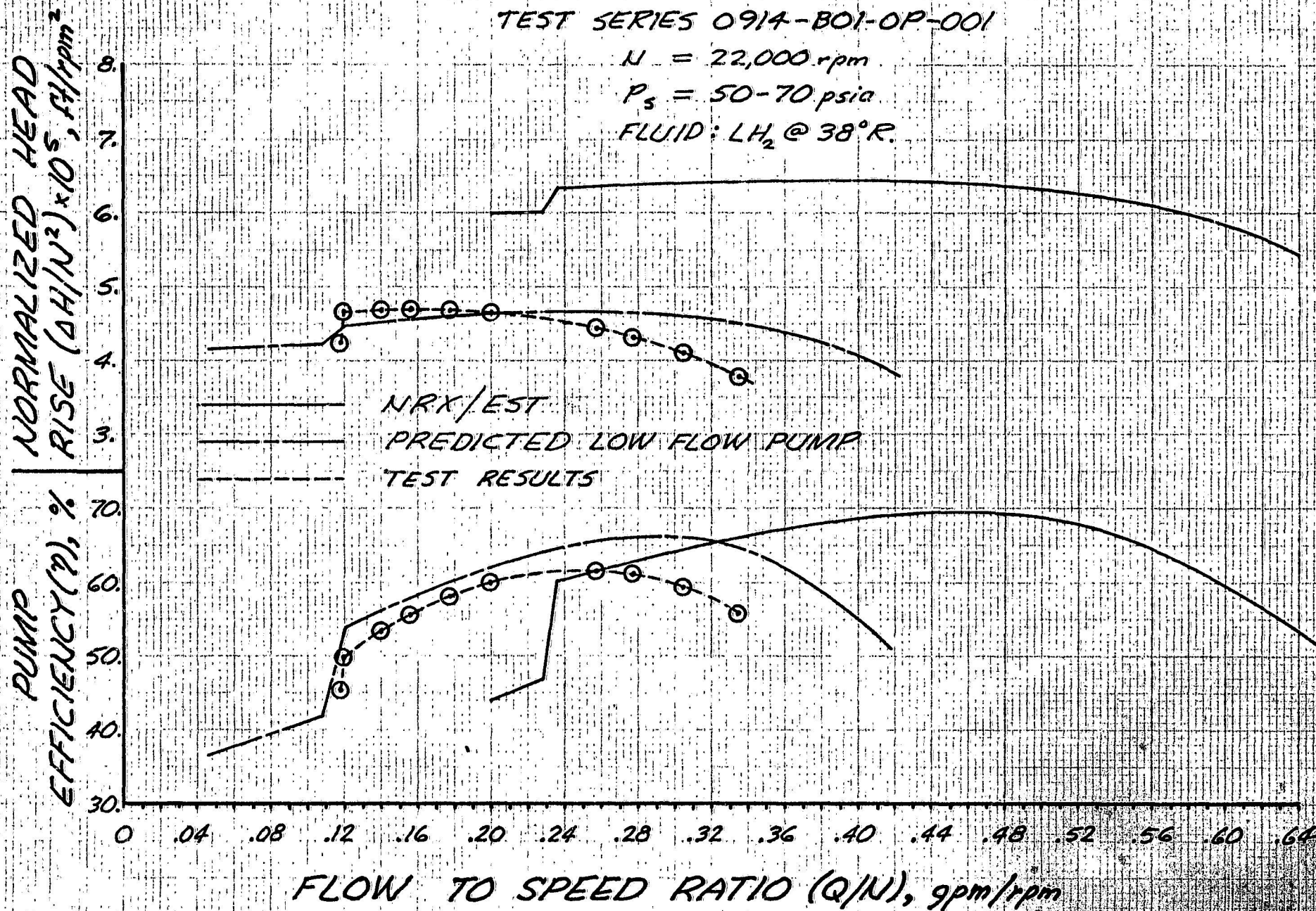


Figure 3
25

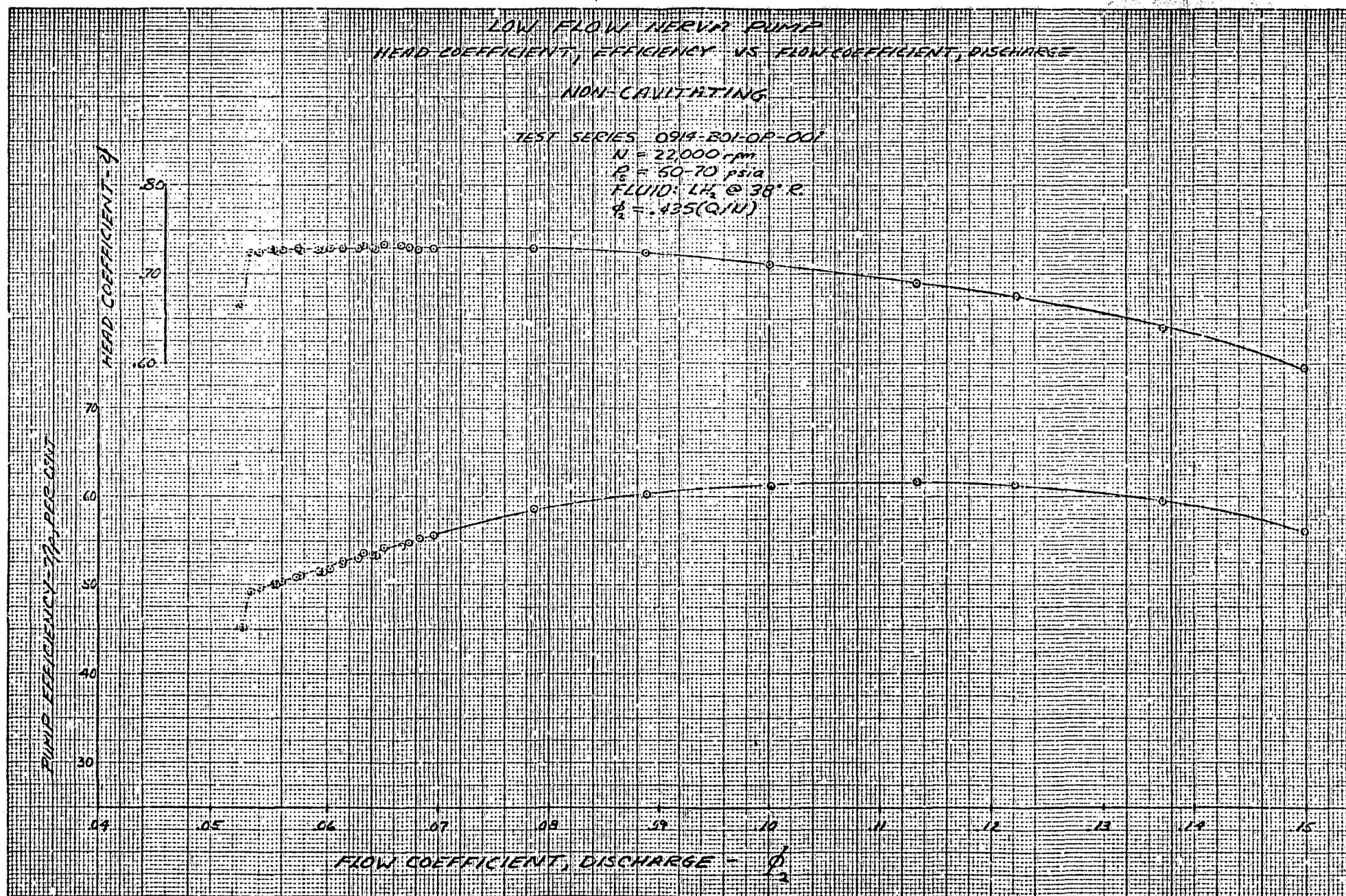


Figure 4
26

NERVA LOW FLOW TECHNOLOGY PUMP NON-CAVITATING PERFORMANCE

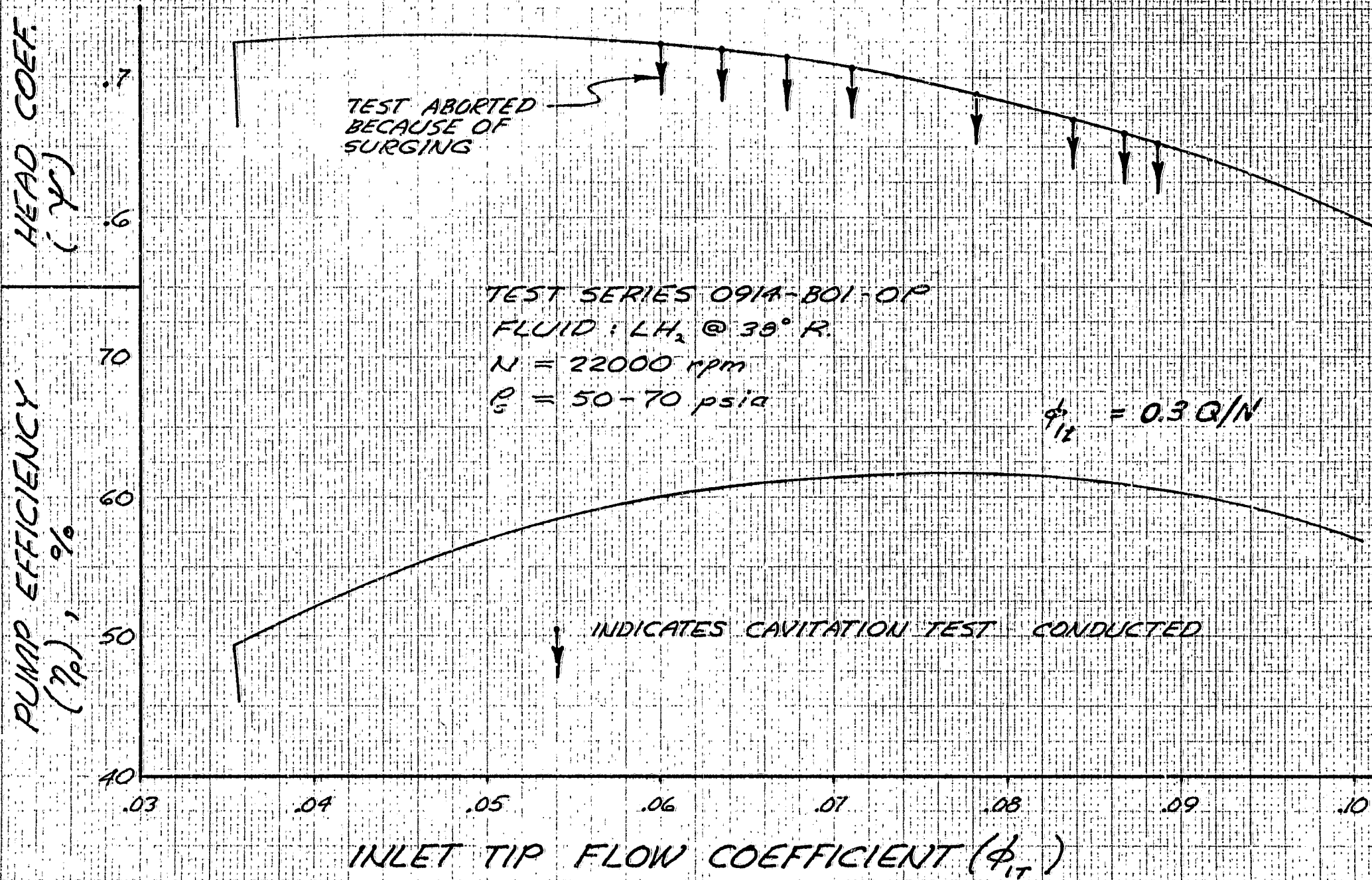
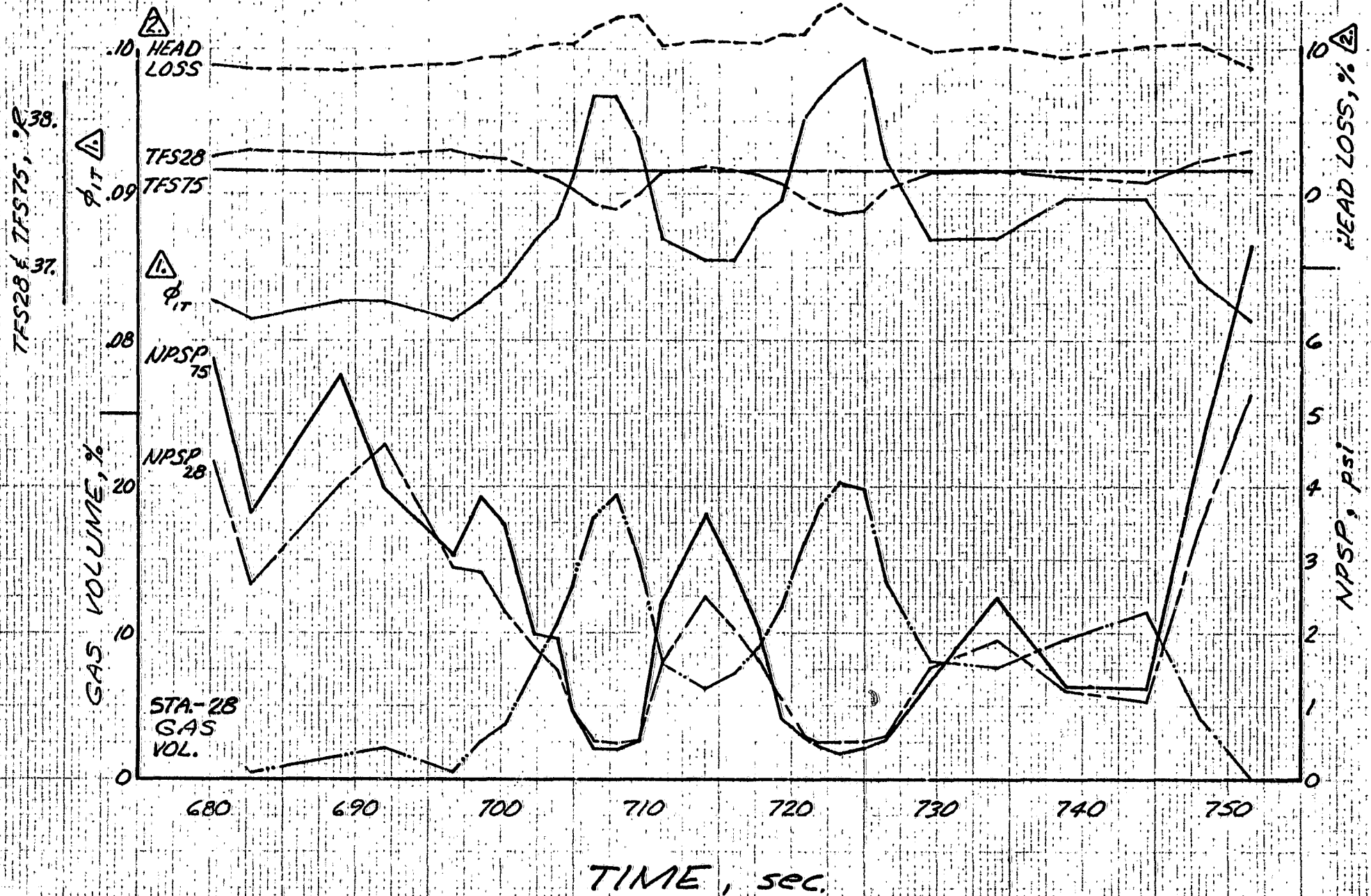


Figure 5
27

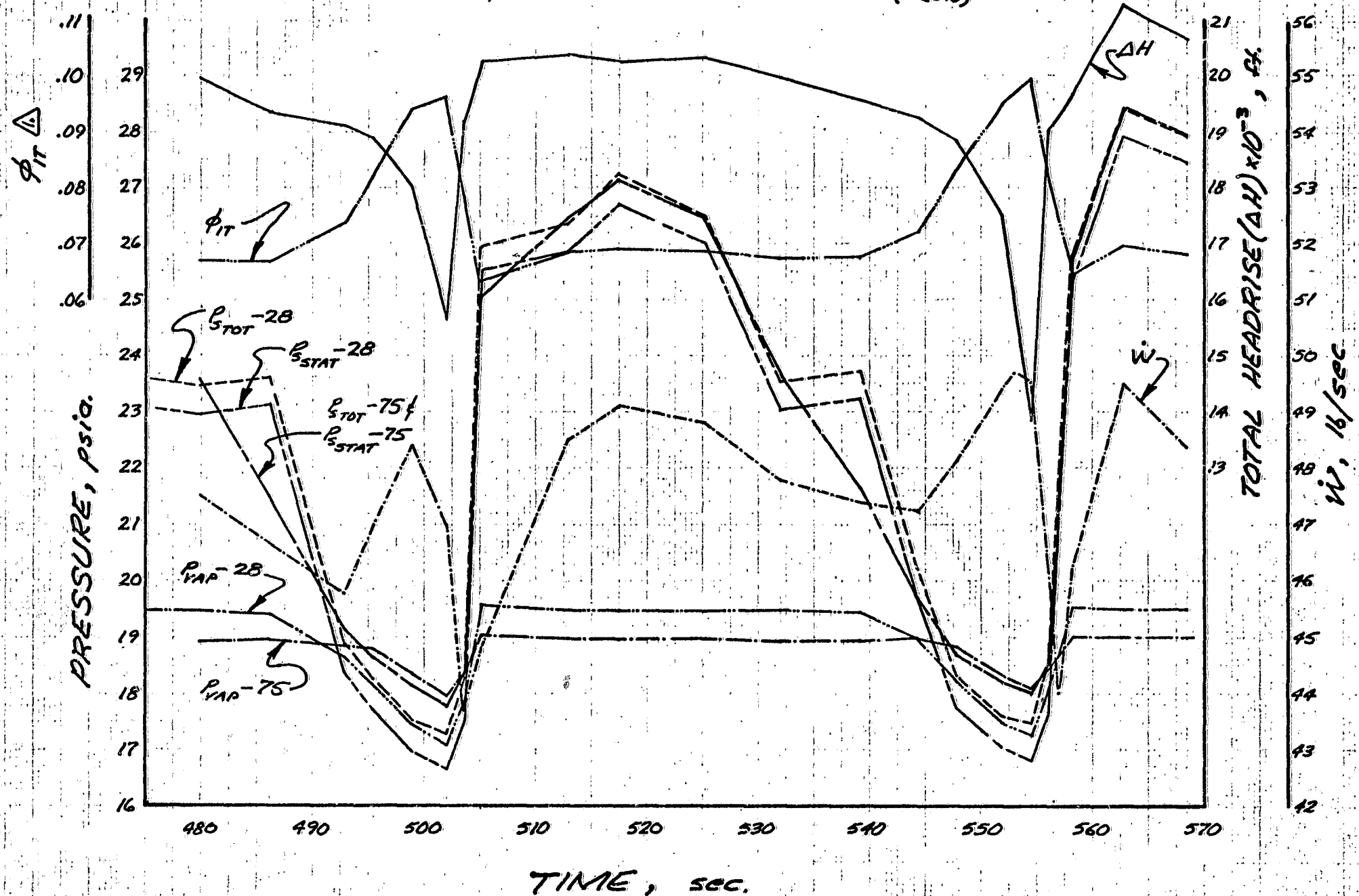
NERVA LOW FLOW PUMP CAVITATION TEST SER. 0914-801-OP-001, TEST NO. 2 $N = 22,000$ rpm $\phi_{IT}(\text{LIQUID}) = .084$



NOTES:

- △ INLET FLOW COEFFICIENT ADJUSTED FOR 2-PHASE FLOW BASED ON FLUID DENSITY AT STA.-28
- △ BASED ON NON-CAVITATING HEADRISE MEASURED AT 50 psi NPSP

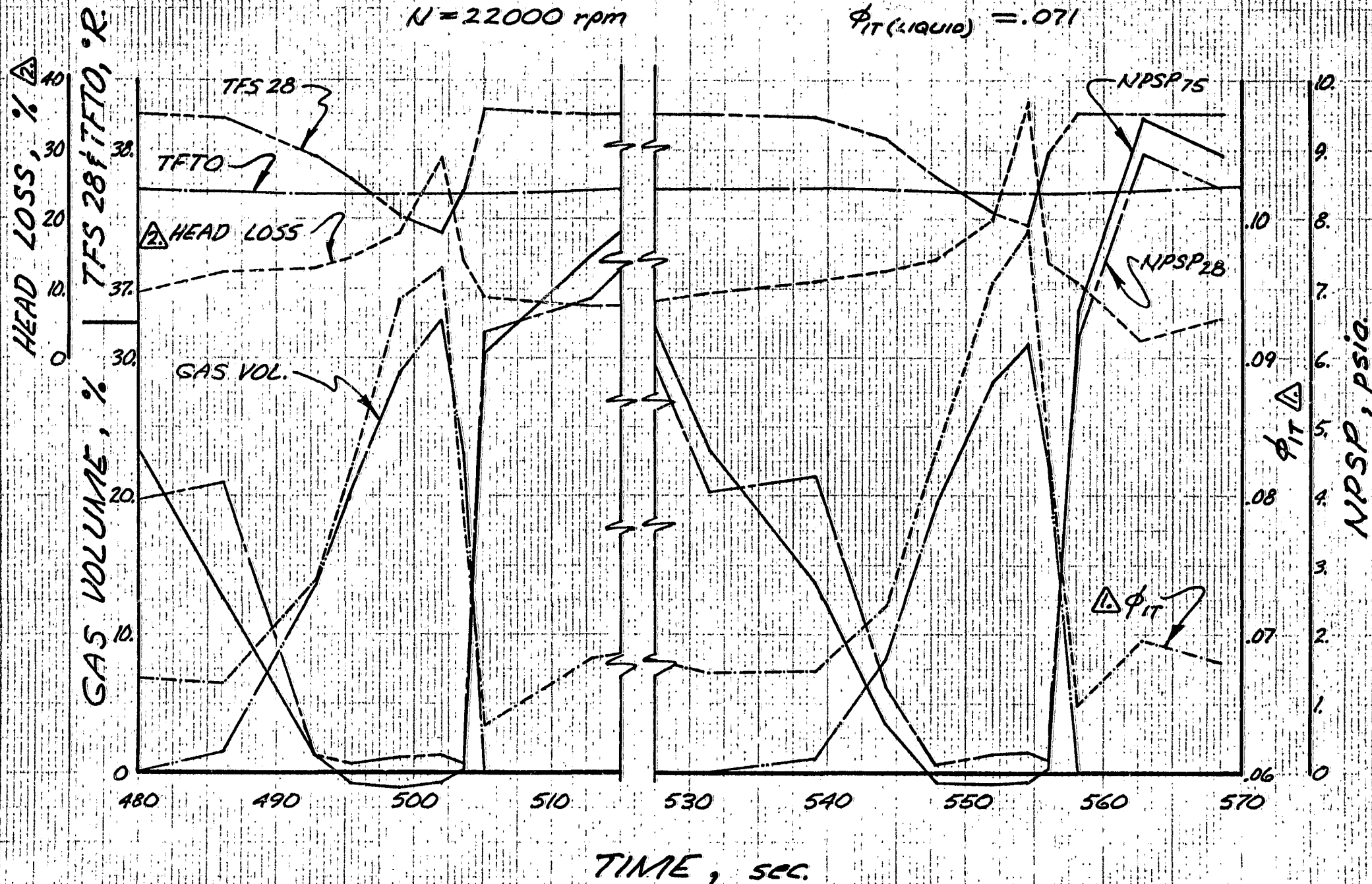
NERVA LOW FLOW TECHNOLOGY PUMP
CAVITATION TEST SER. 0914-801-OP-002, TEST NO. 3
N = 22000 rpm $\phi_{IT}(\text{LIQUID}) = .071$



NOTE

① INLET FLOW COEFFICIENT ADJUSTED FOR 2-PHASE FLOW BASED ON FLUID DENSITY AT STA.-28

NERVA LOW FLOW PUMP
CAVITATION TEST SER. 0914-B01-OP-002, TEST 1103
N = 22000 rpm
 $\phi_{IT}(\text{LIQUID}) = .071$



NOTES:

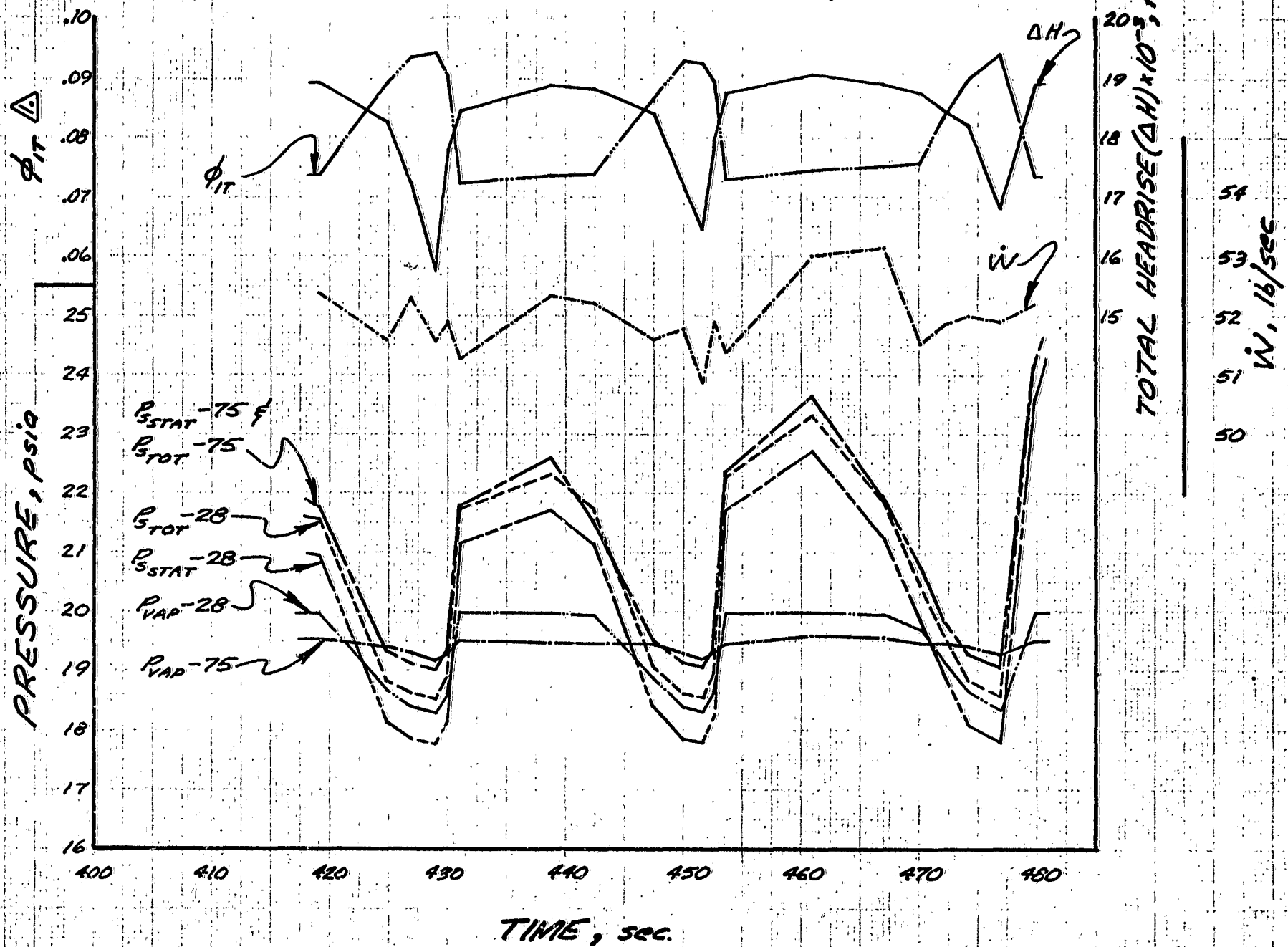
- △ INLET FLOW COEFFICIENT ADJUSTED FOR 2-PHASE FLOW BASED ON FLUID DENSITY AT STA.-28
- △ BASED ON NON-CAVITATING HEADRISE MEASURED AT 50 psi NPSP

NERVA LOW FLOW TECHNOLOGY PUMP

CAVITATION TEST SER. 0914-801-OP-003, TEST NO. 5

$N = 22000 \text{ rpm}$

$\phi_{IT}(\text{LIQUID}) = .078$



NOTE

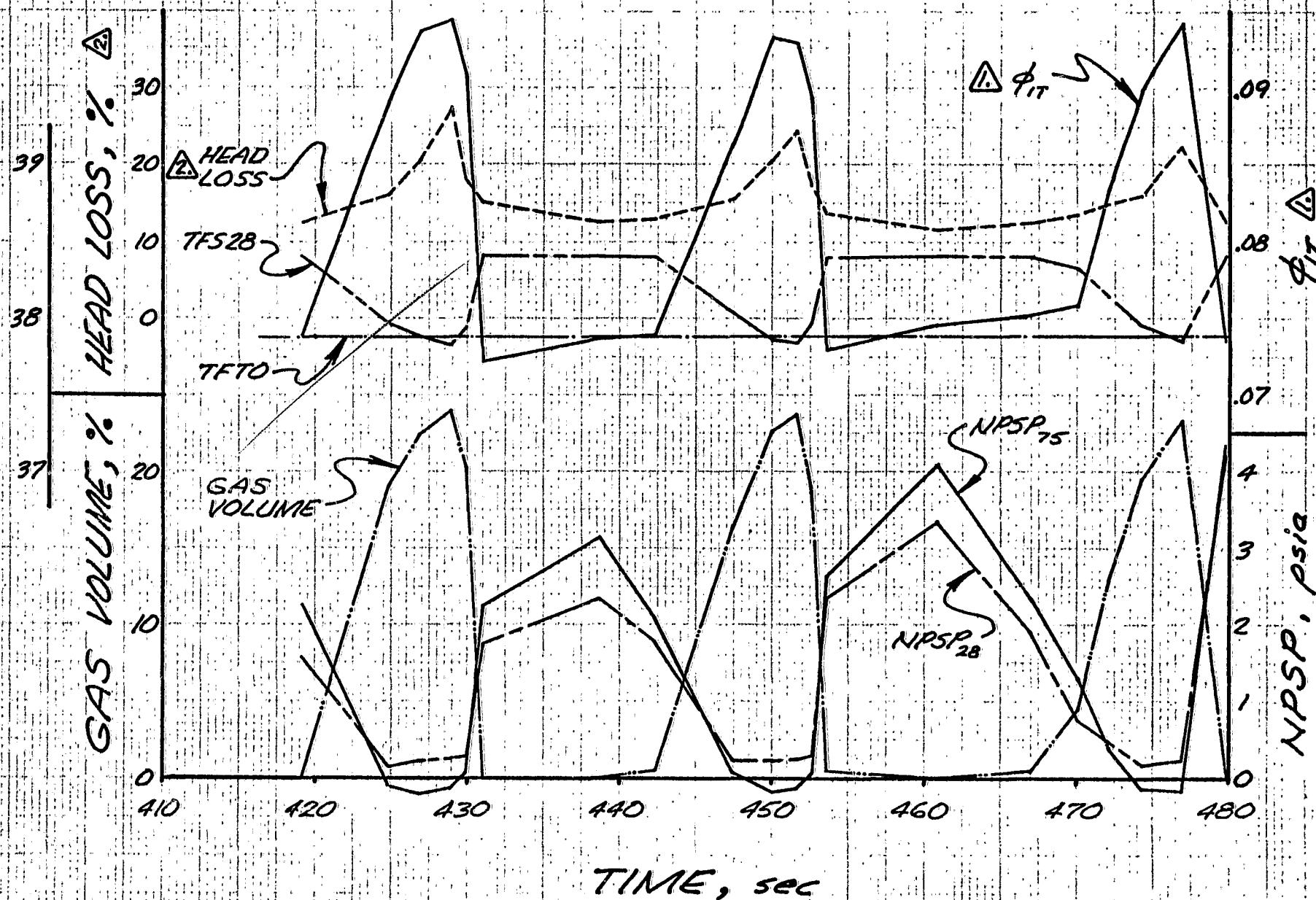
Δ INLET FLOW COEFFICIENT ADJUSTED FOR 2-PHASE FLOW BASED ON FLUID DENSITY AT STA-28

NERVA LOW FLOW TECHNOLOGY PUMP

CAVITATION TEST SER. 0914-B01-OP-003, TEST NO. 5

$N = 22000 \text{ rpm}$

$\phi_{IT}(\text{LIQUID}) = .078$



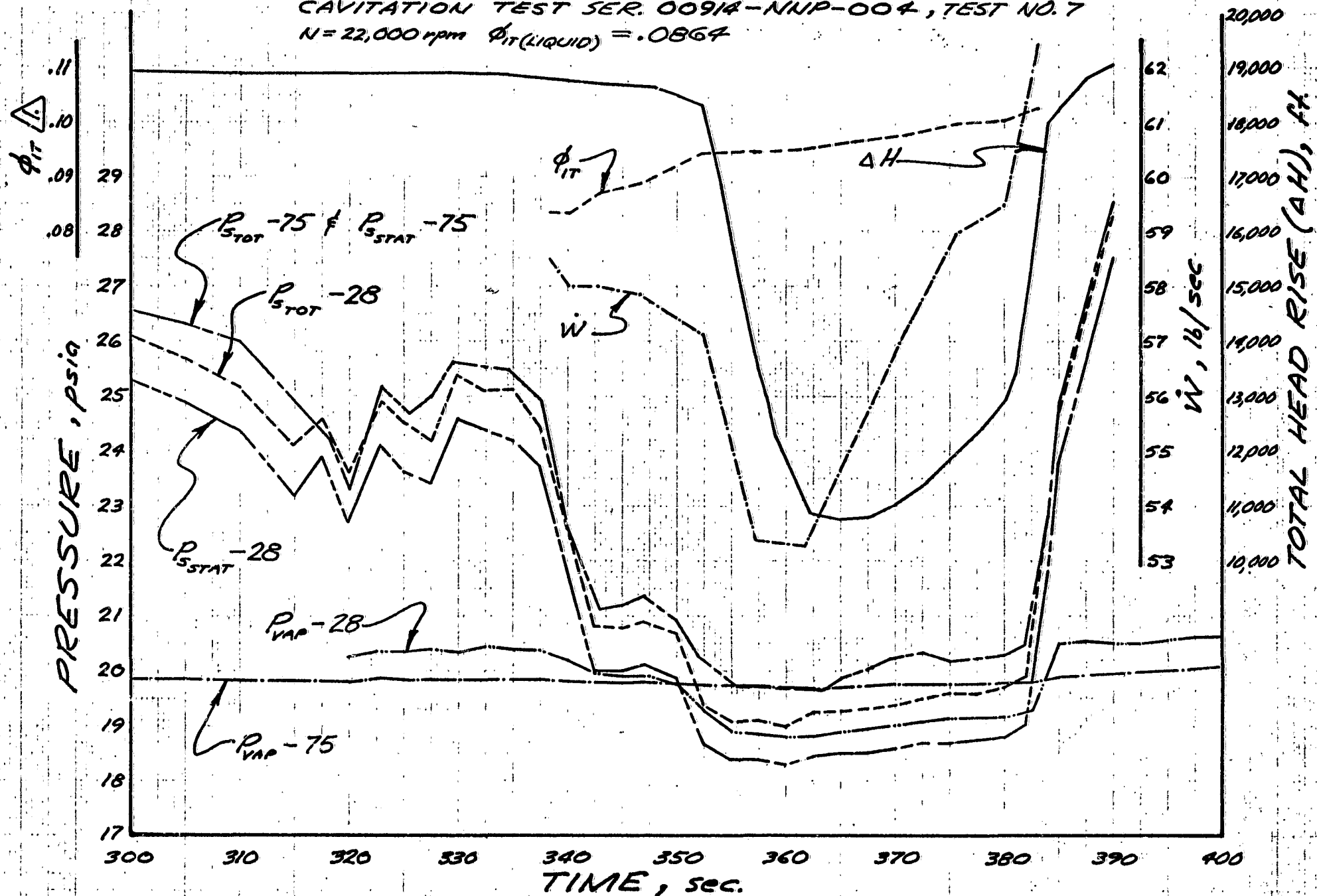
NOTES:

- ① INLET FLOW COEFFICIENT ADJUSTED FOR 2-PHASE FLOW BASED ON FLUID DENSITY AT STA. 28.
- ② BASED ON NON-CAVITATING HEADRISE MEASURED AT 50 psi NPSP.

NERVA TECHNOLOGY TURBOPUMP

CAVITATION TEST SER. 00914-NNP-004, TEST NO. 7

$N = 22,000 \text{ rpm}$ $\phi_{IT}(\text{LIQUID}) = .0864$

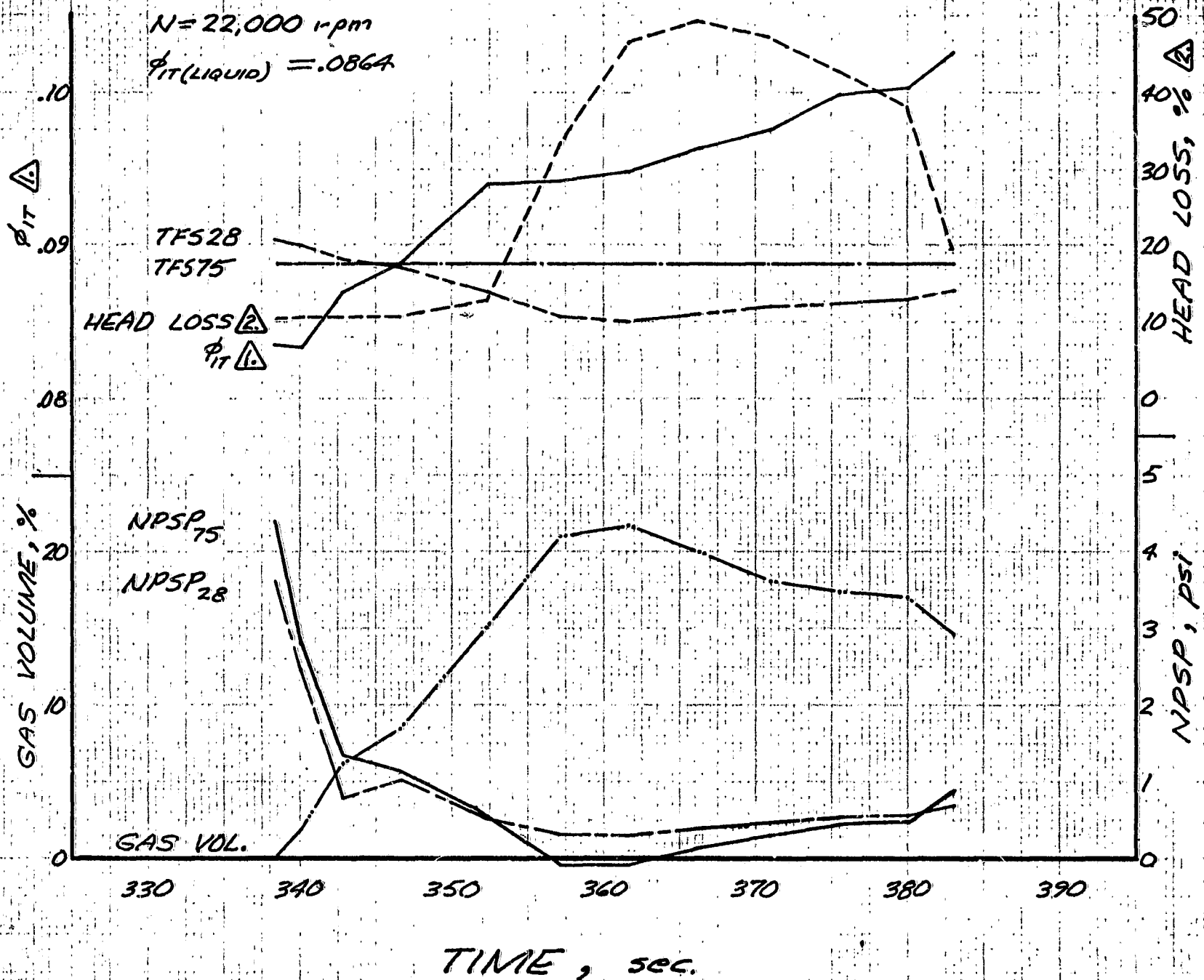


Δ INLET FLOW COEFFICIENT ADJUSTED FOR 2-PHASE FLOW BASED ON FLUID DENSITY AT STATION 28.

NERVA LOW FLOW PUMP CAVITATION TEST SER. 0914-B01-OP-004, TEST NO. 7

$N = 22,000$ r.p.m.
 $\phi_{IT}(\text{LIQUID}) = .0864$

TF575, %R. 39.
TF528, %R. 38.



NOTES:

1. INLET FLOW COEFFICIENT ADJUSTED FOR 2-PHASE FLOW BASED ON FLUID DENSITY AT STA-28
2. BASED ON NON-CAVITATING HEADRISE MEASURED AT 50 psi NPSH

NERVA LOW FLOW TECHNOLOGY PUMP

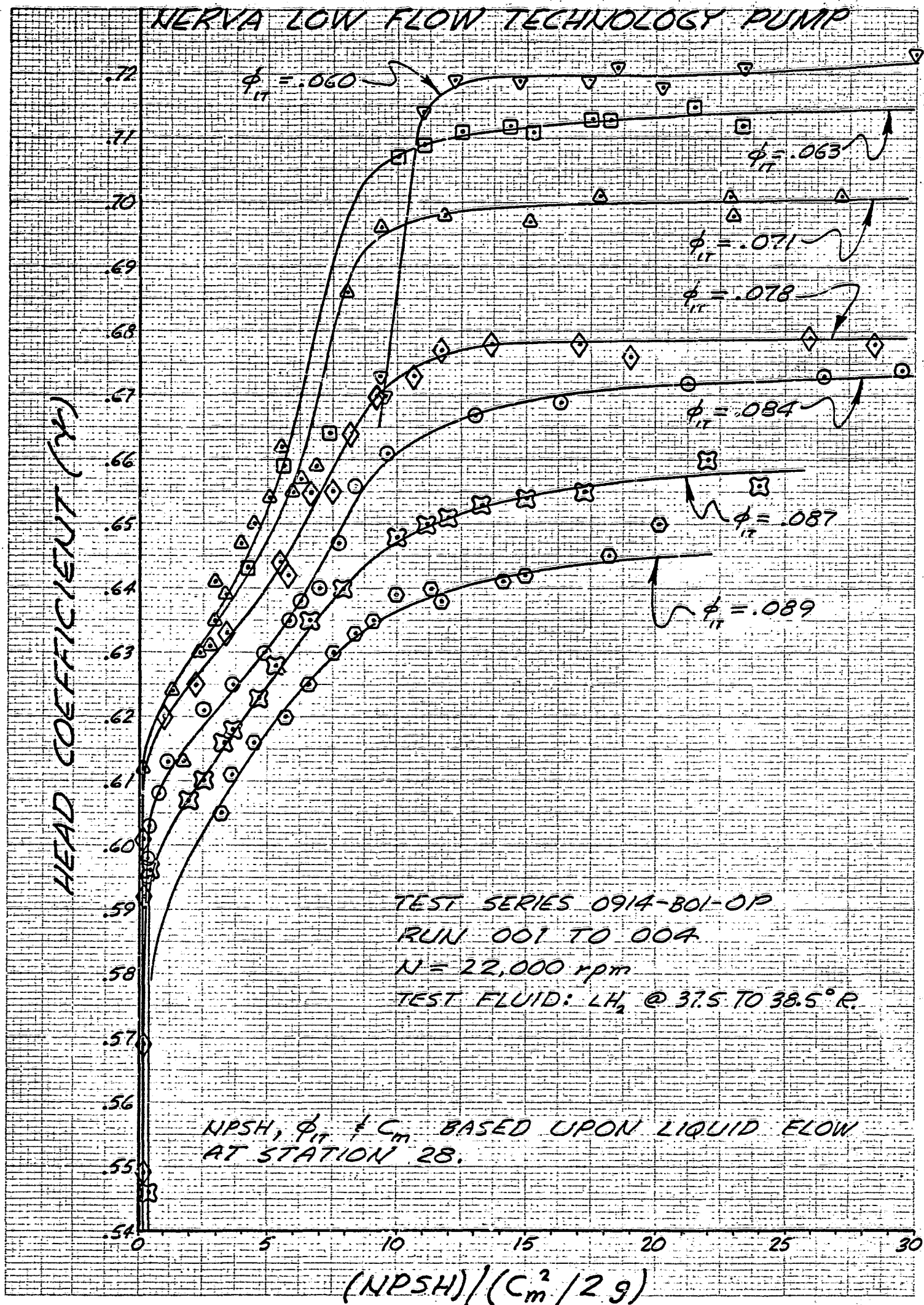
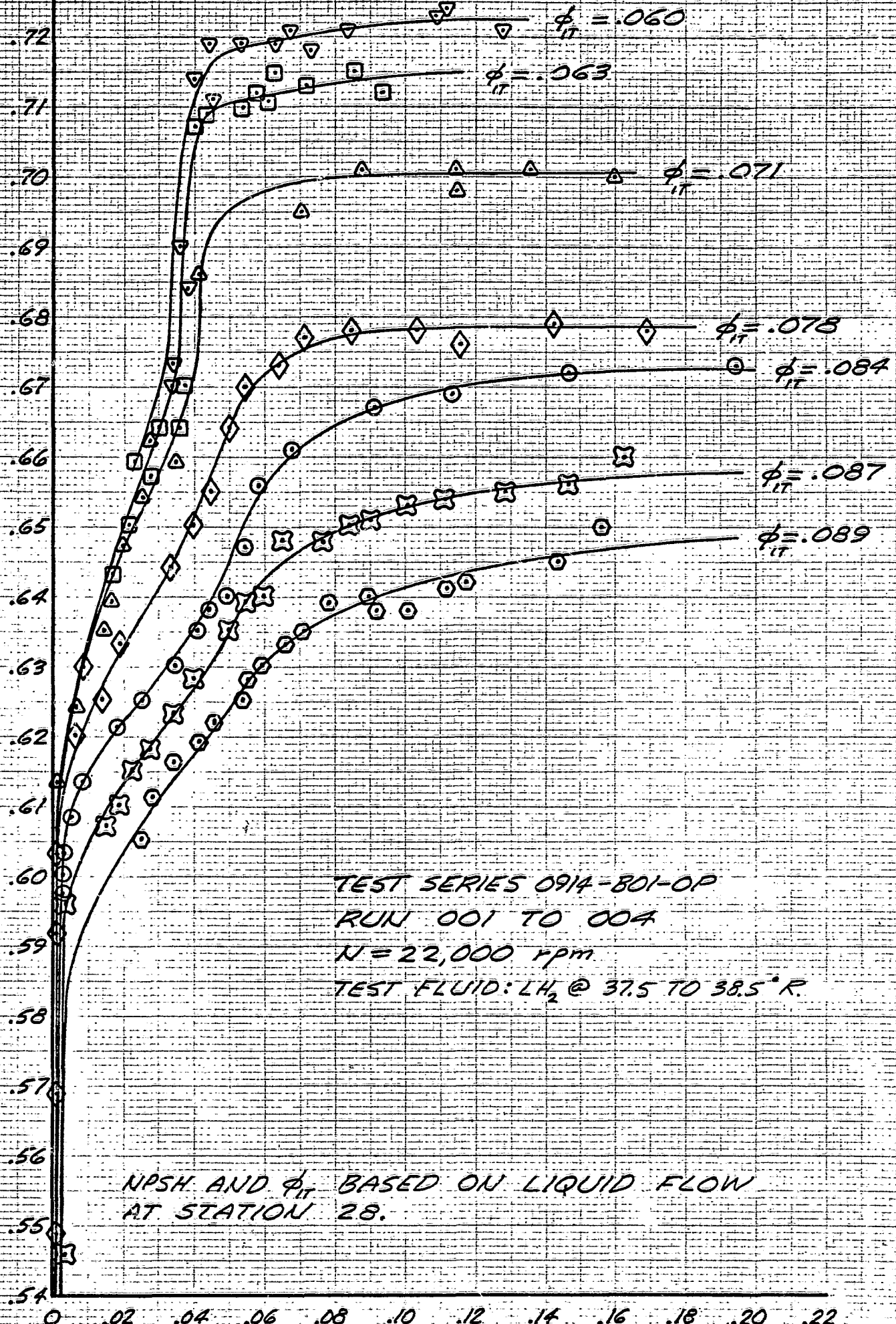


Figure 10
 36

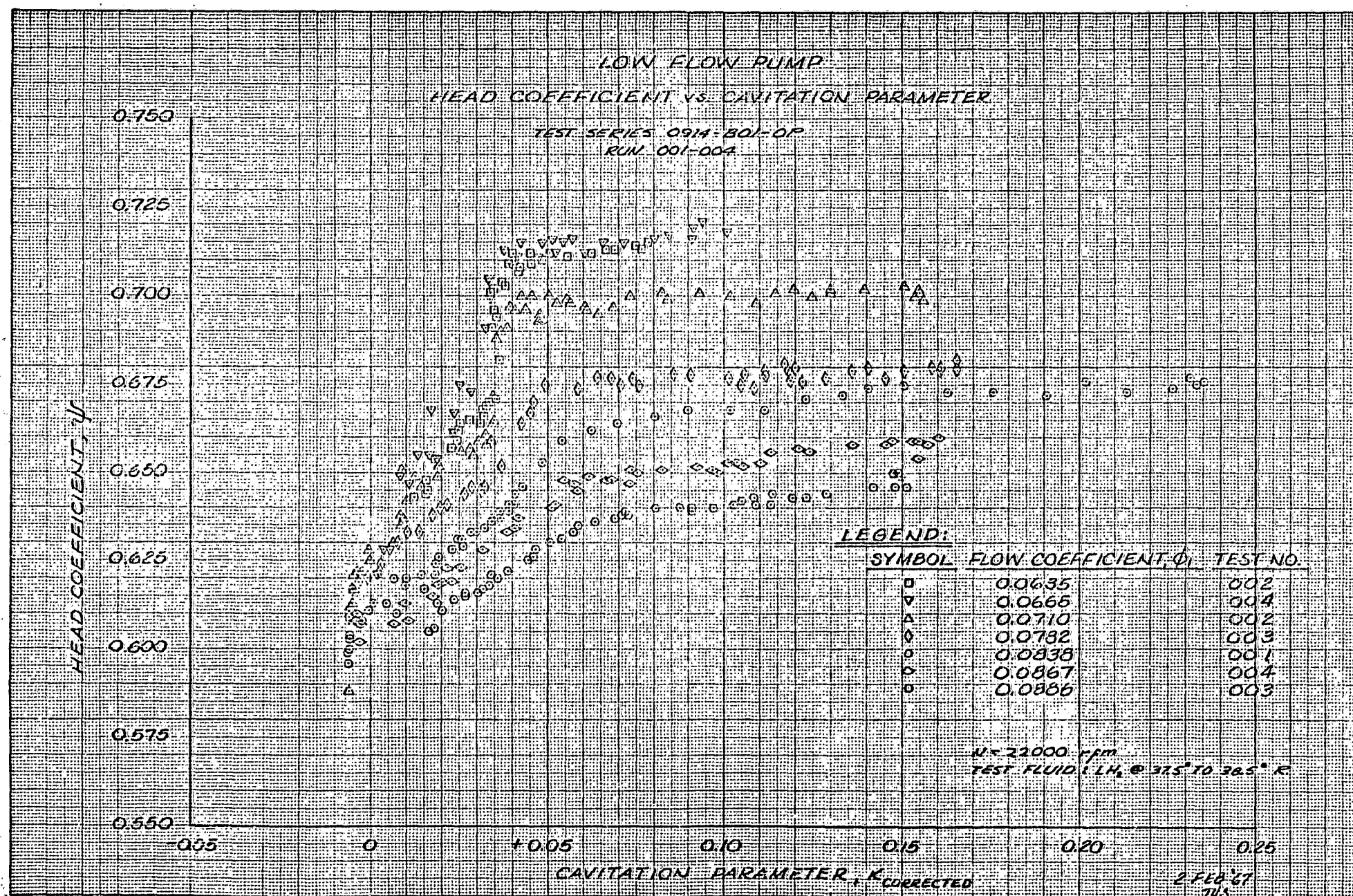
HEAD COEFFICIENT (η)



$$\tau = (NPSH) / (U^2 / 2g)$$

Figure 11
37

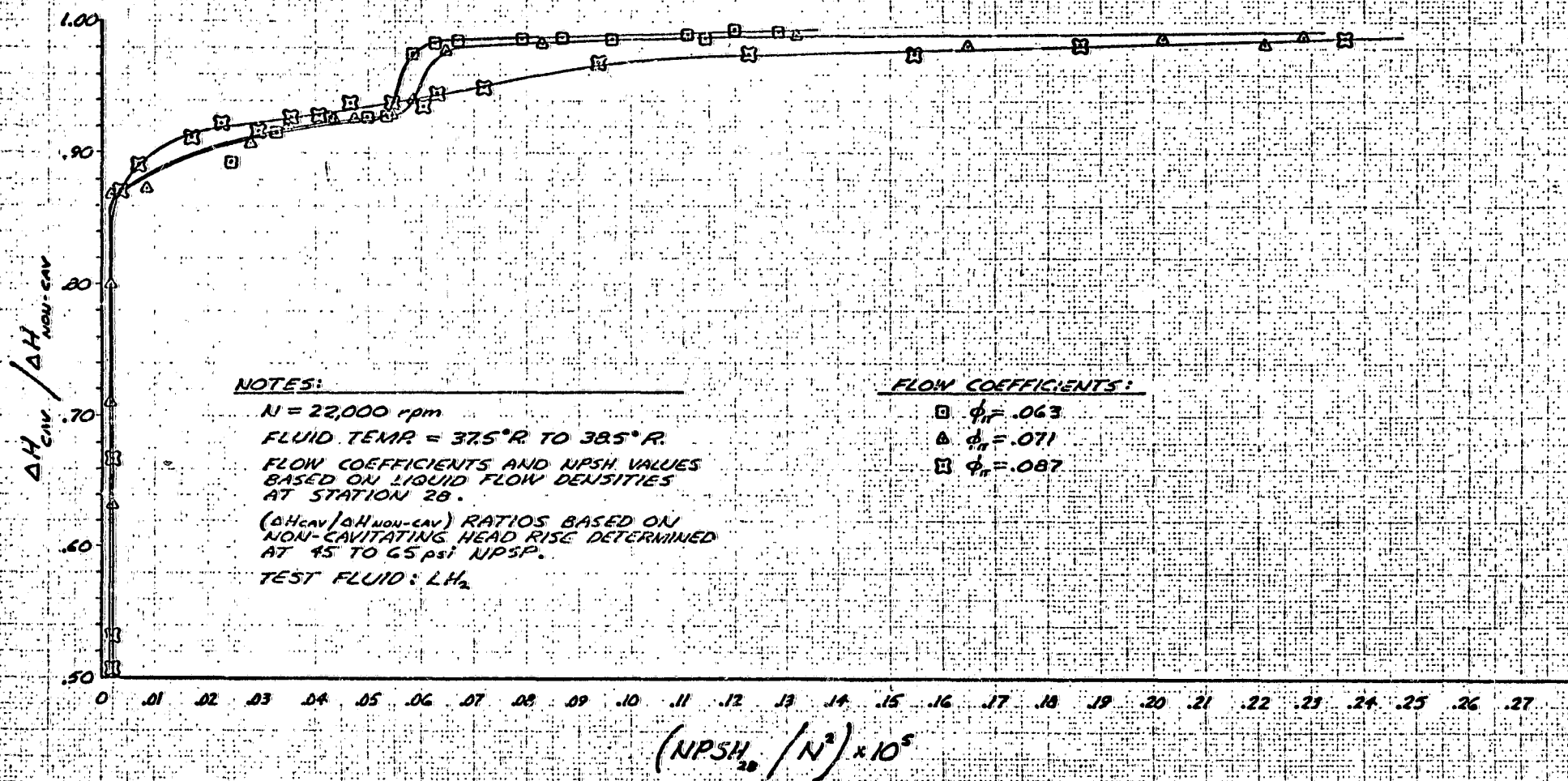
10 X 10 PER HALF INCH
 NO. 240V-1015 DIST. FROM PAPER-11-10-10
 MADE IN U.S.A.
 10 X 10 PER HALF INCH



NERVA LOW FLOW TECHNOLOGY PUMP CAVITATION PERFORMANCE

$$\frac{\Delta H_{\text{CAV}}}{\Delta H_{\text{NON-CAV}}} \text{ VS } \frac{\text{NPSH}_{28}}{N^2}$$

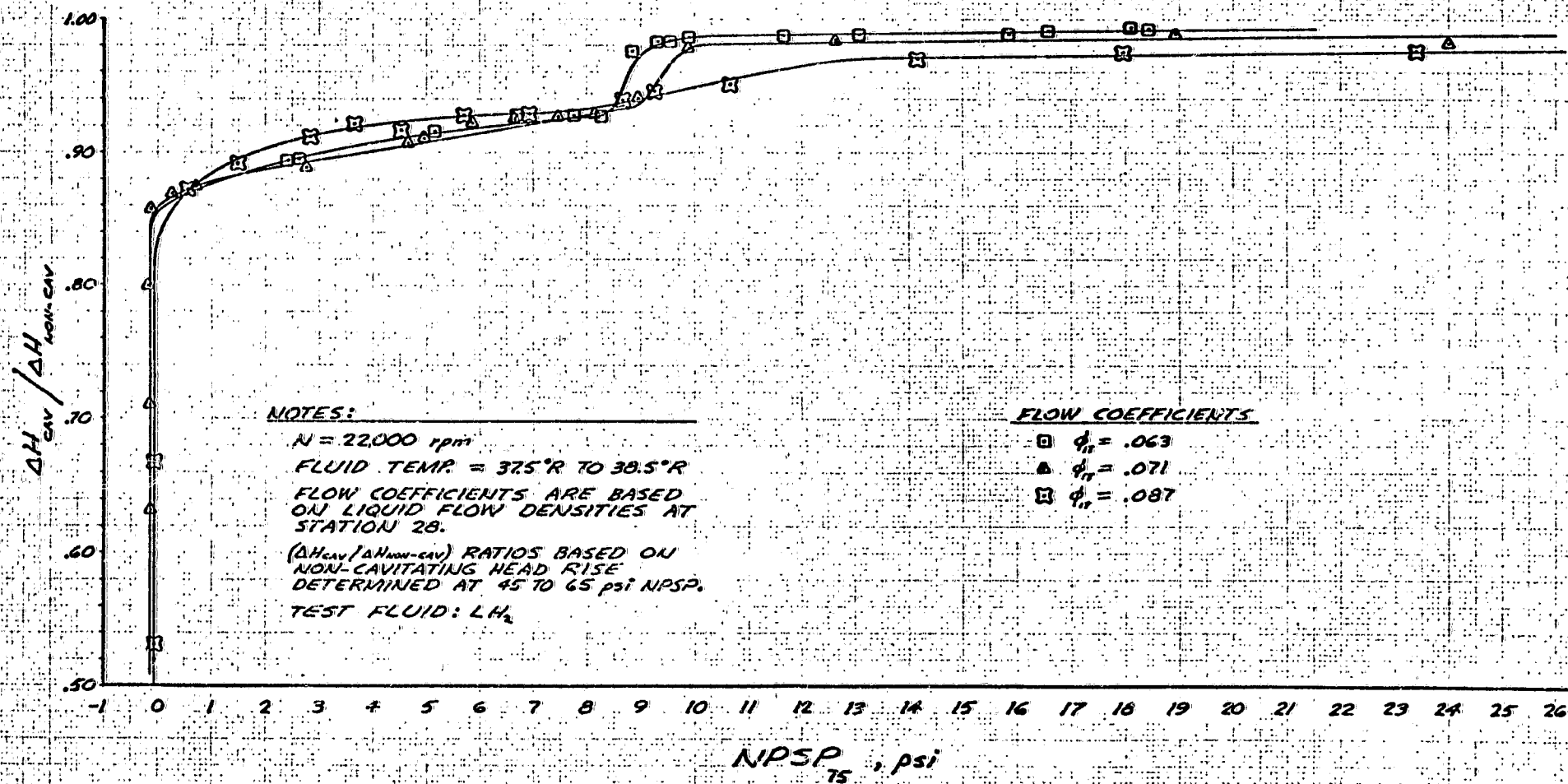
TEST SERIES 0914-B01-OP
RUN 001 TO 004



NERVA LOW FLOW TECHNOLOGY PUMP CAVITATION PERFORMANCE

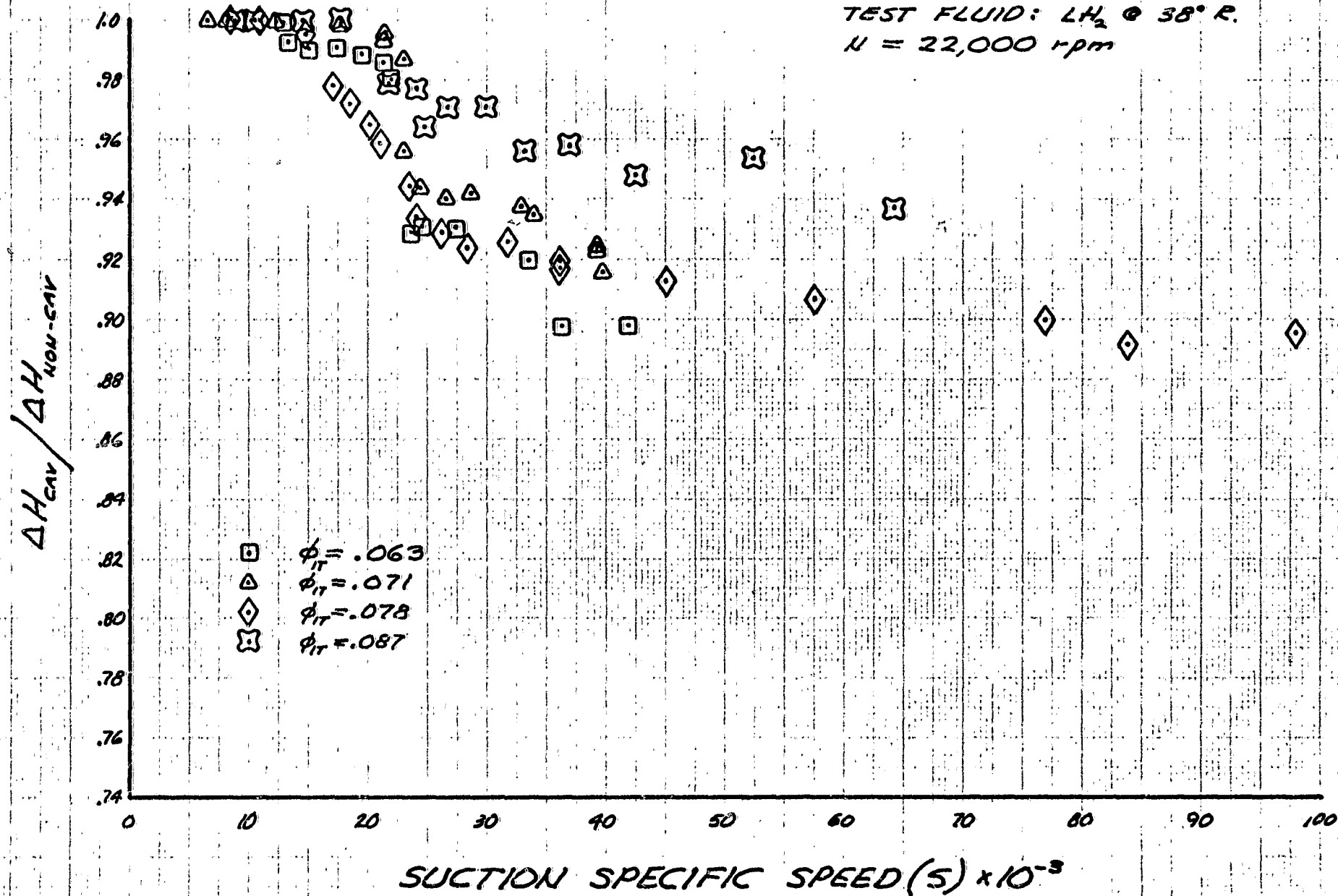
$$\frac{\Delta H_{\text{cav}}}{\Delta H_{\text{non-cav}}} \text{ vs } \text{NPSP}_{75}$$

TEST SERIES 091A-B01-OP
RUN 001 TO 004



NERVA LOW FLOW TECHNOLOGY PUMP $\Delta H_{CAV} / \Delta H_{NON-CAV}$ vs SUCTION SPECIFIC SPEED

TEST SERIES 0914-B01-OP
 TEST FLUID: LH_2 @ $38^\circ R$.
 $N = 22,000$ rpm



NERVA LOW FLOW TECHNOLOGY PUMP
CAVITATION PARAMETER (τ) vs
INLET TIP FLOW COEFFICIENT (ϕ_{IT})
AT 3, 7, & 10% HEAD LOSS

TEST SERIES: 0914-B01-0P

TEST FLUID: LH₂ @ 38° R.

N = 22,000 rpm

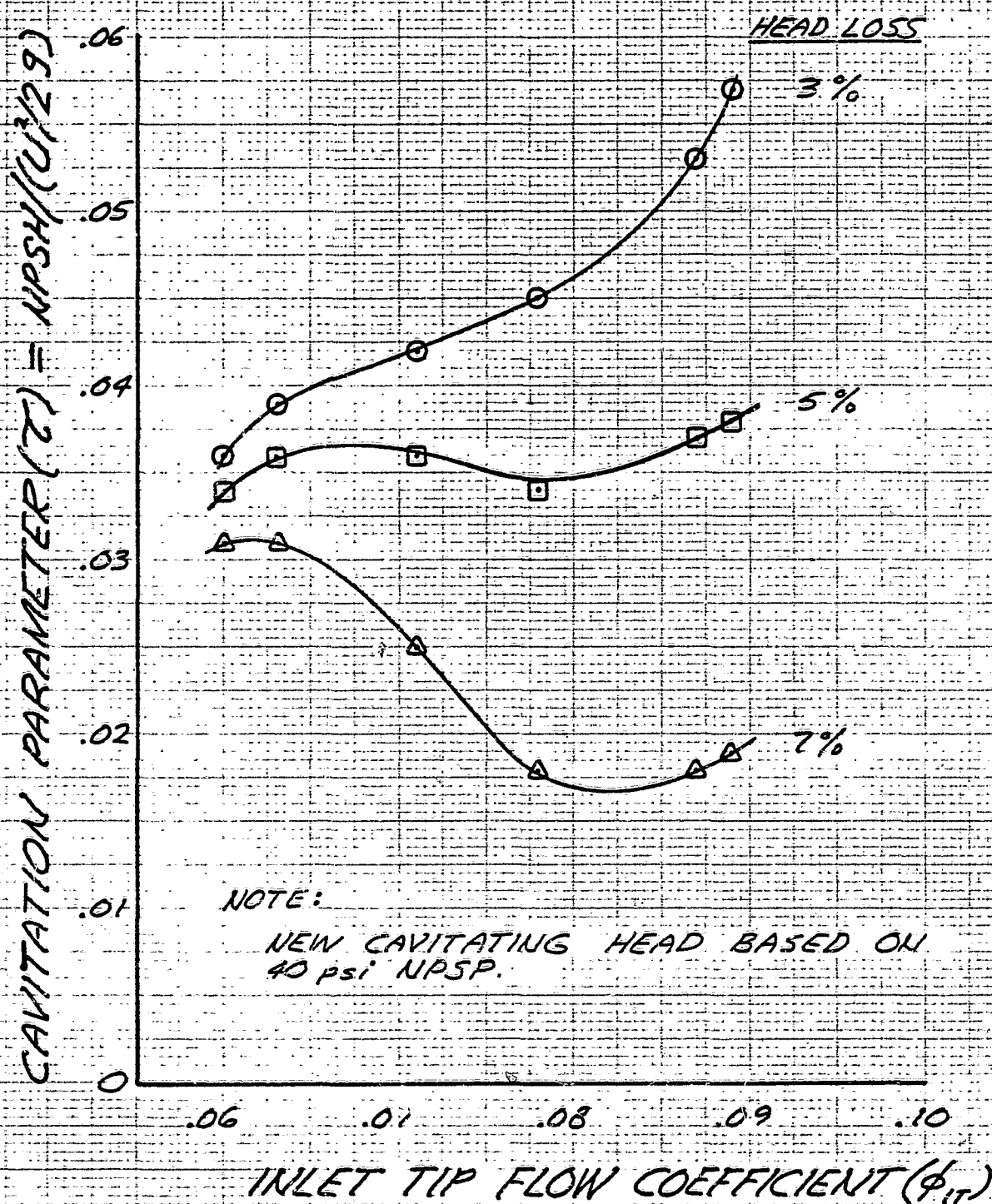


Figure 16

1. 5 IN., 6° NASA FLAT PLATE INDUCER $U_{it} = 436$ FT/SEC
2. MODIFIED NERVA TECHNOLOGY PUMP RADIAL IMPELLER WITH INTEGRAL INDUCER
 $\beta_{it} = 7^\circ$, $D_{it} = 7$ IN., $U_{it} = 670$ FT/SEC
3. 5 IN., 11° NASA CAMBERED INDUCER $U_{it} = 435$ FT/SEC

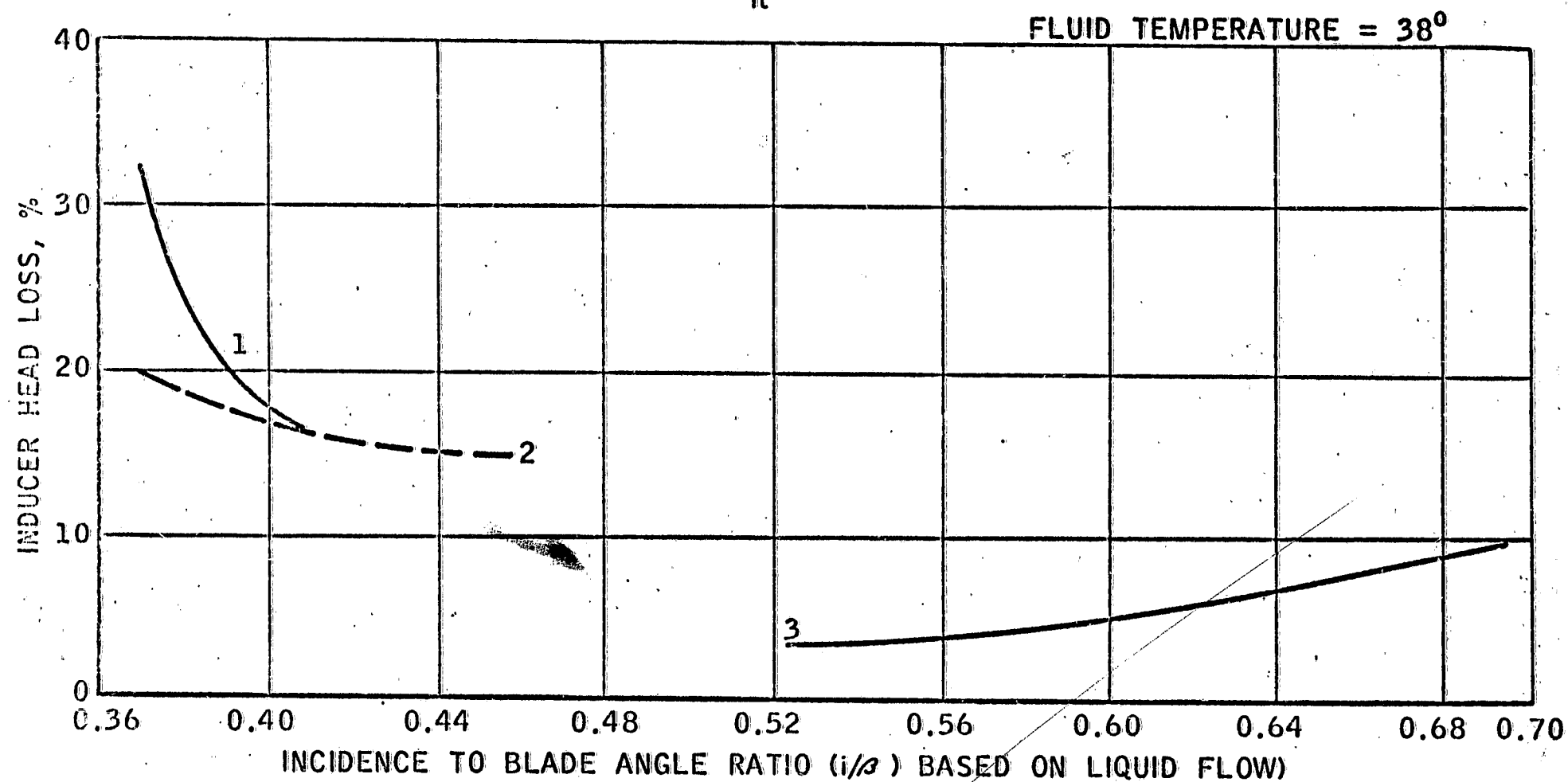


Figure 17
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NERVA
LOW FLOW PUMP
CAVITATION PERFORMANCE
CAVITATION PARAMETER(K) $\leq$ INLET FLOW COEFFICIENT(ϕ_{IT})
DATA FROM TEST SERIES 0914-B01-0P
3% HEAD LOSS
TEST FLUID: LH₂ @ 38°R.

NOTE:

HEAD LOSS BASED
UPON NON-CAVITATING
DATA TAKEN AT 45 TO
65 psi NPSP DURING
TEST 1.

ASC EXPERIENCE
BOUNDARIES

$$K = (NPSP - C_m^2/2g) / (W_r^2/2g)$$

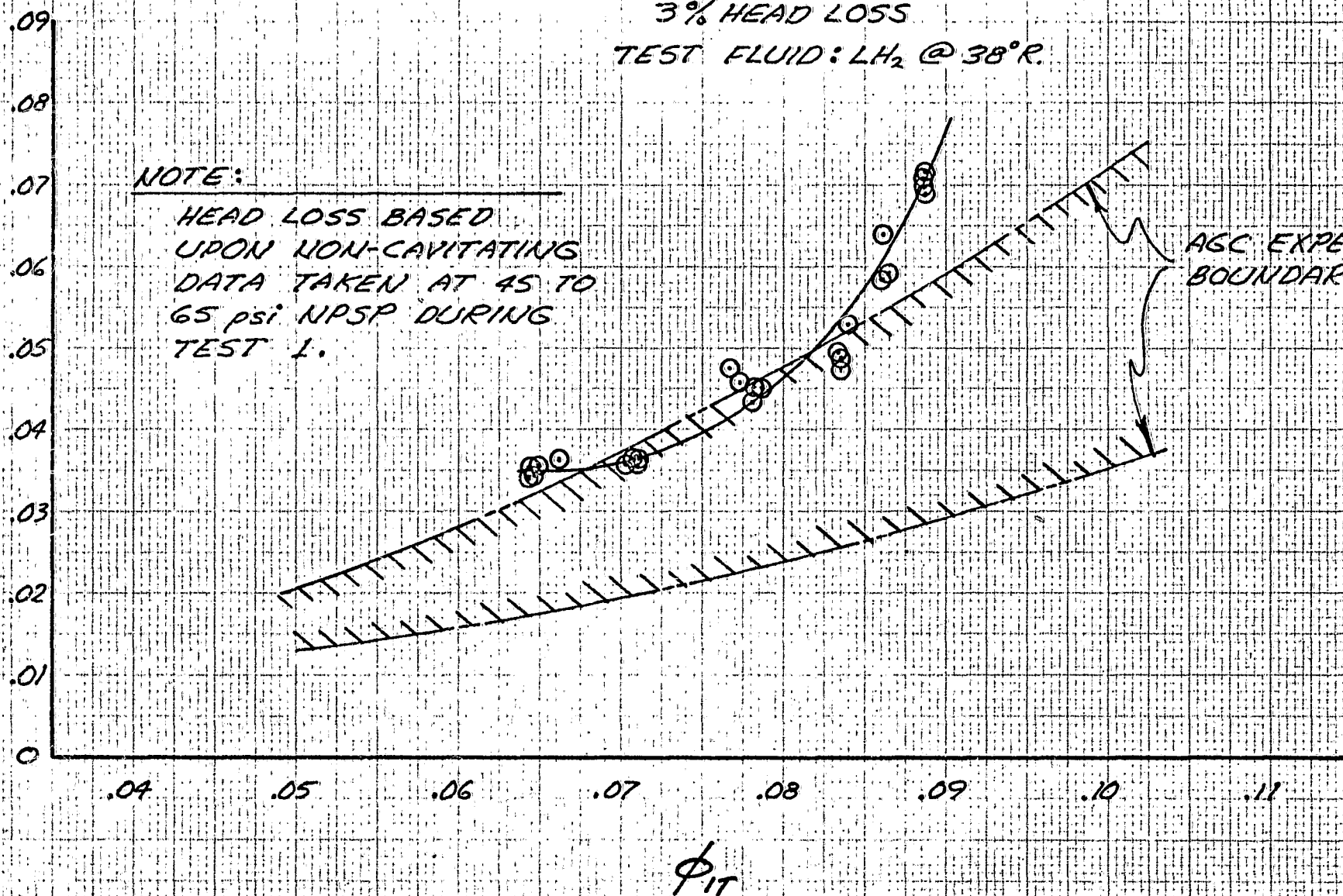
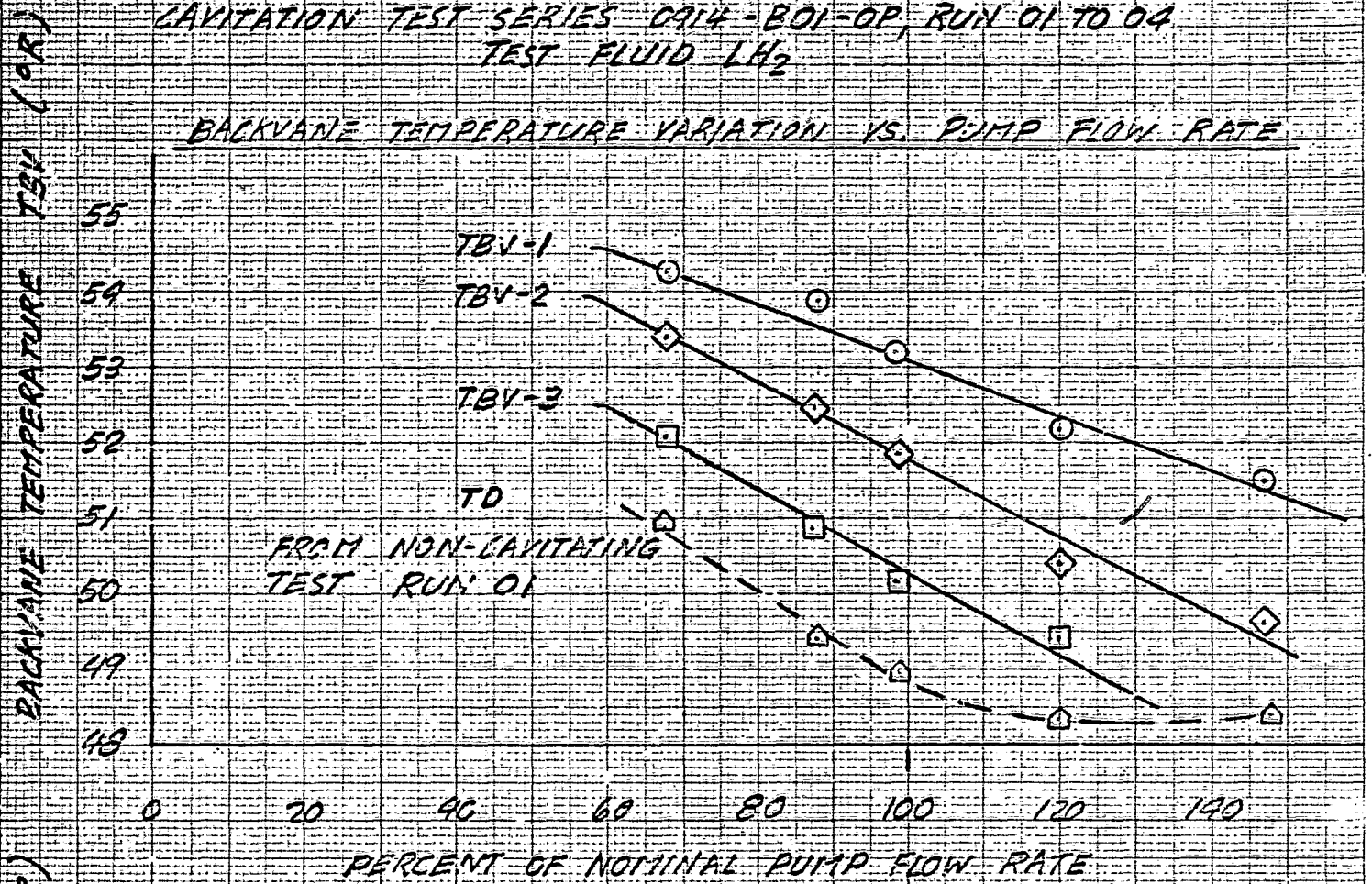


Figure 18
44

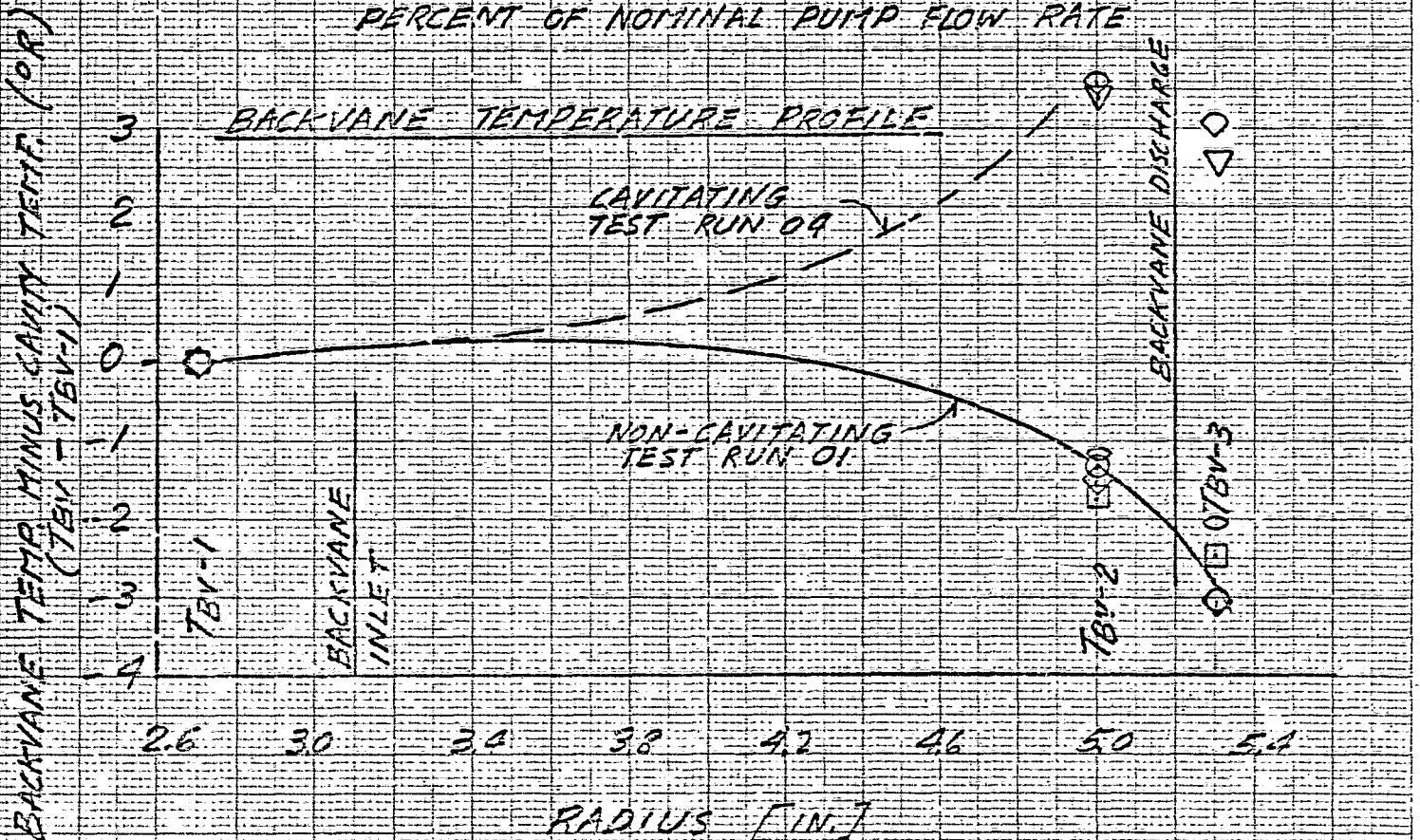
BACKVANE TEMPERATURE EVALUATION

CAVITATION TEST SERIES 0914-B01-OP, RUN 01 TO 04
TEST FLUID LH₂

BACKVANE TEMPERATURE VARIATION VS. PUMP FLOW RATE



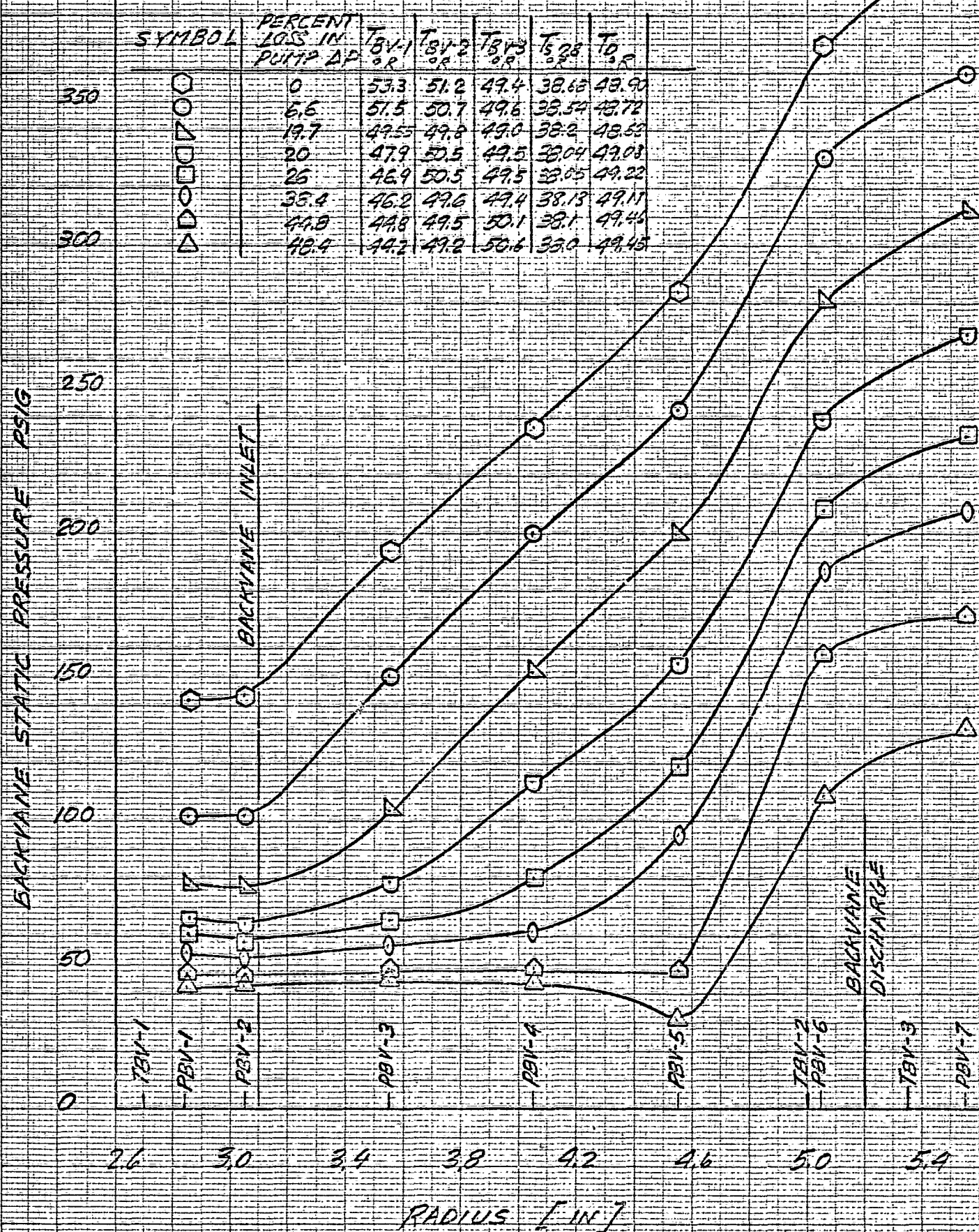
BACKVANE TEMPERATURE PROFILE



6-8-67 7/2

Figure 19

BACKVANE PRESSURE PROFILES AT VARIOUS DEGREES OF PUMP CAVITATION
RUN 4, TEST 7, TEST FLUID LH₂



6-13-67 J.F.H.

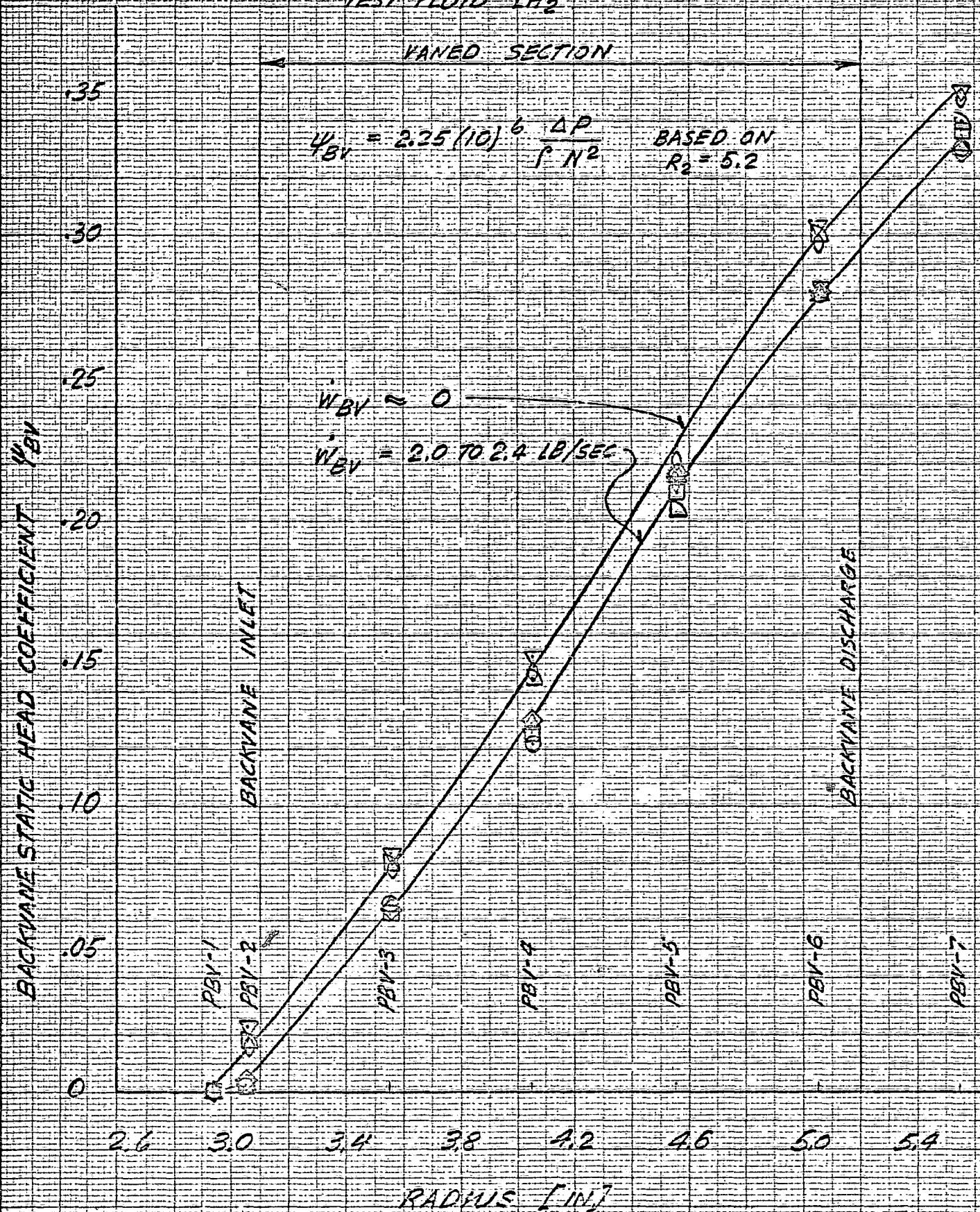
Figure 20
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NERVA LOW FLOW TECHNOLOGY PUMP

NON-CAVITATING BACKVANE PERFORMANCE

TEST SERIES 0914-B01-OP, RUN 01 TO 04

TEST FLUID LH₂



6-8-67 *21*

Figure 21

REFERENCES

1. Devita, V. A.; Hydrogen Fluid Temperature Measurements in NERVA Nozzle Coolant Tubes Using Thermocouples, AGC Plant Test Operations Engineering Report No. 9830-52, 7 October 1965.
2. Brunner, J. J.; Analysis and Experimental Verification of Axial Thrust on the M-1 Liquid Oxygen Turbopump, NASA CR 54817.
3. Brunner, J. J.; M-1 Oxidizer Scale Pump Performance with Initial and Interim Design Impellers, Aerojet Report TPR 0047, April 1966.
4. Pratt & Whitney Aircraft, Florida Research and Development Center, Investigation of Pressure Prediction Methods for Radial Flow Impellers, PWA-FR-1276, 8 March 1965.